SLIDE VALVE GEARS.

AN EXPLANATION OF THE ACTION AND CONSTRUCTION OF PLAIN AND CUT-OFF SLIDE VALVES.

BY

FREDERICK A. HALSEY,

ASSOCIATE EDITOR OF THE "AMERICAN MACHINIST."

CONSULTING ENGINEER OF THE RAND DRILL COMPANY; MEMBER OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

Analysis by the Bilgram Diagram

SIXTH EDITION, REVISED AND ENLARGED.

NEW YORK:

D. VAN NOSTRAND COMPANY,

23 MURRAY AND 27 WARREN STREETS.

1899.

Printed by
Braunworth, Munn & Barber
Brooklyn, N. Y., U. S. A.

TO

Professor John E. Sweet,

TO HAVE BEEN WHOSE PUPIL I CONSIDER ONE OF THE GREATEST PRIVILEGES OF MY LIFE,

THIS LITTLE VOLUME IS GRATEFULLY INSCRIBED.

PREFACE.

THIS work has been prepared to meet what the author considers a real want. It has been written with the aim of making it intelligible to any one who might be willing to make a serious effort to understand it. High authority exists for a mathematical treatment of the subject, but with this the author has no sympathy. Designing a valve gear is essentially a drawing board process, and a mathematical treatment of it is simply an uncalled for use of heavy artillery. The graphical treatment is therefore adopted throughout.

Acknowledgment is due to Mr. Hugo Bilgram for his courtesy in kindly permitting the use of his valve diagram. The author has all due respect for the Zeuner diagram, but that respect is not incompatible with the conviction that Mr. Bilgram's method is a marked improvement upon it. Valve diagrams are used for two purposes—to analyze existing valve motions and to design new ones. The Zeuner diagram fulfils the first purpose perfectly, but is unsatisfactory when applied to the second. The leading data that are given in designing a valve motion are the point of cut-off, the port opening, and the lead of the valve (not the lead angle of the crank, as is often conveniently assumed). It is the radi-

cal defect of the Zeuner diagram that none of these dimensions can be laid off from known points. The lead must be laid off from an unknown point of the centre line, and the port opening from an unknown point on an unknown line. Finally, through these unknown points and the centre of the shaft the valve circle is to be drawn from an unknown centre and with an unknown radius. Under these circumstances the result sought is found only through blind trial. With Mr. Bilgram's method all this is changed. The lead is laid off from a fixed line, the port opening from a fixed point, and the cut-off position of the crank is located. The lap circle is then drawn tangent to these lines, and the problem is solved. Moreover, the awkward conception of the backward rotation of the crank is obviated. Finally, these marked advantages are not accompanied by any compensating disadvantages whatever.

Acknowledgment is also due to the *American Machinist* for the use of a number of engravings originally prepared to illustrate some of the author's articles in that paper.

The irregularities due to the connecting rod introduce peculiar difficulties into the study of the first principles of the slide valve, which difficulties were first overcome by the happy expedient of using the slotted cross-head instead of the connecting rod in the preliminary study. For this, together with many other original and highly valuable contributions to the subject, we are indebted to Mr. W. S. Auchincloss, who first published them in his well-known and standard work entitled *Link and Valve Motions*, to which those who wish to prosecute their studies beyond the scope of this work are referred.

The author has gone more fully than is customary into the methods of equalizing the various events of the stroke. The sections relating to these methods will be found more difficult to follow than the others, while at the same time they form no necessary part of a *general* treatment of the subject. Those who begin their studies of valve motions with this book, may find these chapters too difficult for the first reading. They have, therefore, been marked with a star (*) in the Table of Contents and in the body of the book, in order that they may be omitted, if desired, in the first reading; and it should be understood that the chapters not so marked form of themselves a complete connected treatise, of a more elementary character than the book as a whole.

PHILADELPHIA, Oct. 19, 1889.

PREFACE TO THE SIXTH EDITION.

THE chief addition to this edition will be found in Part IV, which comprises some articles written for the *American Machinist* and republished here by permission of the American Machinist Publishing Co.

The analysis of the action of the link motion here given, while qualitative rather than quantitative, is believed to recognize two sources of error in that gear for the first time, and to show that the error due to the angular vibration of the connecting rod, heretofore considered the chief error of the gear, is really of minor importance, and, in fact, properly considered, is a corrective and not a disturbing factor, since its effect is to partially compensate another and much larger error.

Advantage has been taken of the opportunity offered, to make a few minor changes, the most important of which is the extension of the section upon Exhaust Lap, with the addition of two full page diagrams to it. The author's observations have shown him that much confusion prevails among students regarding this subject, which seemed to justify a more detailed treatment of it than that given in previous editions.

The author desires to acknowledge his obligations to the Schenectady Locomotive Works for facilities afforded for examining their present-day practice in link motion construction, without which the study here embodied in print would not have been undertaken.

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PART I.

THE SLIDE VALVE WITH FIXED ECCENTRIC.

THE SLIDE VALVE WITH FIXED ECCENTRIC.

THE PLAIN SLIDE VALVE.

Fig. 1 is a sectional view of a plain slide valve and its seat, the valve being shown in its central position, with the ports completely covered by it. The distance

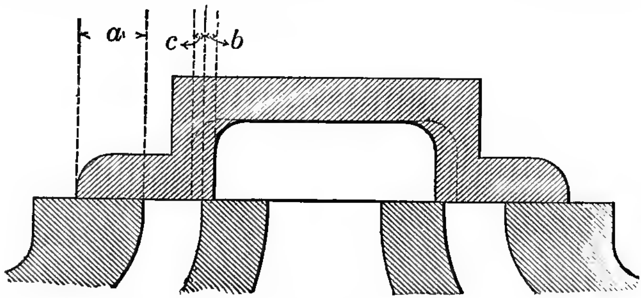

Fig. 1

a, by which the valve extends beyond the steam edge of the port, is called the outside lap, steam lap, or more usually, simply lap of the valve. The distance *b*, by which it extends beyond the exhaust edge of the port, is called the inside lap or exhaust lap.* The exhaust

* As will be more fully shown later on, valves are sometimes so made that the steam is admitted by the inside and exhausted by the outside edges. Hence the terms inside and outside lap are somewhat ambiguous. The terms steam lap and exhaust lap avoid this ambiguity, and are to be preferred.

lap is always much smaller than the steam lap. It is frequently absent, and frequently the exhaust edge of the valve does not reach the exhaust edge of the port, being made as shown by the dotted lines. In that case the distance *c* is usually called inside clearance, though a better name is negative inside or exhaust lap. It is sometimes called inside lead or exhaust lead; but these terms should not be applied here, as they have properly another definite meaning, which will be explained farther on. The measurement for both steam and exhaust lap is made for one end of the valve only. Thus if a valve is said to have $\frac{3}{4}$ inch lap, the meaning is that it has that much at each end.

THE ECCENTRIC.

The slide valve is usually driven by means of an eccentric on the crank shaft, and it becomes necessary at the outset to obtain a clear conception of the motion which the eccentric gives. In brief, the eccentric is a short crank with a large crank pin. It is obvious that the motion of a cross-head would not be changed by increasing the size of the crank pin. If a crank pin were enlarged until the crank shaft came within the circumference of the pin, the result would be an eccentric. The arm of a crank is the distance from the centre of the shaft to the centre of the crank pin, and similarly the "throw" of an eccentric is the distance from the centre of the shaft to the centre of the eccentric disc. Usually the centre of the disc is within the circumference of the shaft; but this does not alter

the nature of the device, which remains simply a short crank with a large crank pin.

THE SCOTCH YOKE OR SLOTTED CROSS-HEAD.

The usual method of connecting the cross-head to the crank pin by means of a connecting rod introduces certain distortions and irregularities into the relative motions of the piston and crank. These will be more fully explained farther on, but it is desirable in the first instance to avoid the necessity for considering them, as they greatly complicate the subject. This is accomplished by considering in the first instance an engine having the piston and crank connected by means of the device called the Scotch yoke or slotted cross-head, since that connection is without the distortions mentioned.* As has been explained, the eccentric is essentially a crank; and it follows that the distortions which are introduced by the connecting rod into the motions of the piston and crank, are also introduced by the eccentric rod into the motions of the valve and eccentric. The reasons which lead to the adoption of the slotted cross-head in place of the connecting rod also require its use in place of the eccentric rod. An engine fitted with slotted cross-heads is illustrated in Figs. 2–11. The slotted cross-head will be recognized at once, and is too familiar a device to need further description.

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* The slotted cross-head is employed here with the permission of Mr. W. S. Auchincloss, to whom the thanks of the author are due.

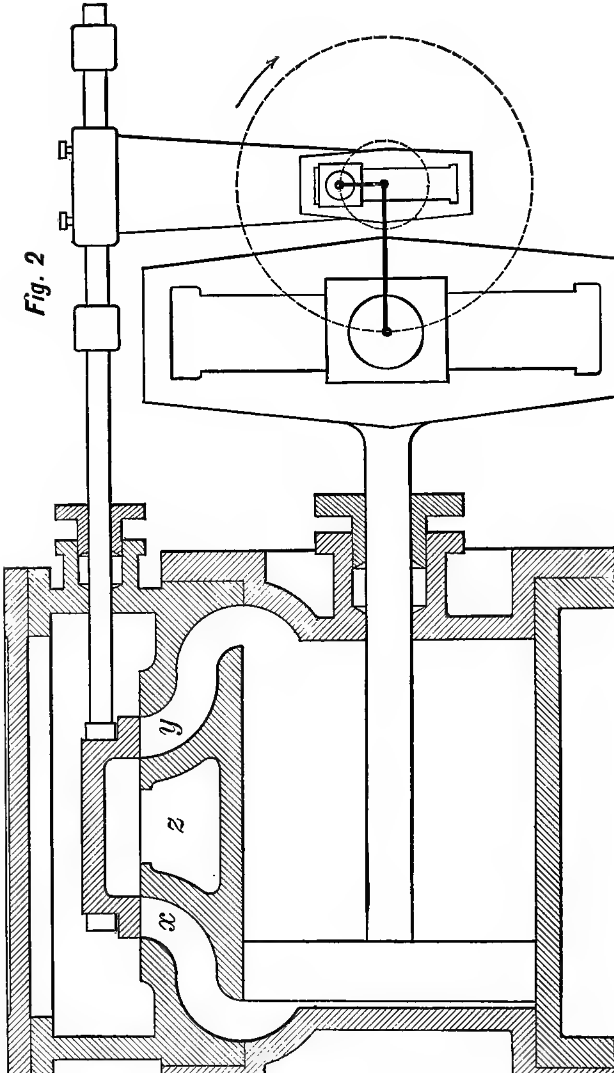


Fig. 2

THE PRIMITIVE ENGINE.

When illustrating the action of the valve of a steam engine, it is essential for clearness that the valve be shown on the top of the cylinder. A valve so located in an actual engine would require the use of a rock shaft to communicate the motion of the eccentric rod to the valve rod—a construction which finds use in American locomotives. This rock shaft complicates the action of the parts, and it is desirable in this preliminary work to avoid it. To accomplish this the unmechanical arrangement of Figs. 2–11 is adopted.

Figs. 2–6 represent the primitive engine with the parts in a number of successive positions. The valve has no lap on either steam or exhaust side, and the eccentric is set at right angles to the crank, and in advance of it in the direction of the rotation. The eccentric, being in fact a crank, is represented as such, and the valve is driven from it by a slotted cross-head, which is secured to the valve stem by the bracket shown. In Figs. 3–6 the slotted cross-heads are represented by their centre lines only, for greater clearness and simplicity.

Referring to Fig. 2, the crank is on the "centre," and the parts are ready to begin movement, the direction of rotation being as shown by the arrow. In Fig. 3 the crank shaft has turned through an angle of forty five degrees, carrying the parts to the positions shown. Considering Figs. 2 and 3, it is clear that the first movement of the crank shaft carried the valve to the right, and thereby opened port x to steam and y to exhaust. Opening port x admitted steam behind the piston to

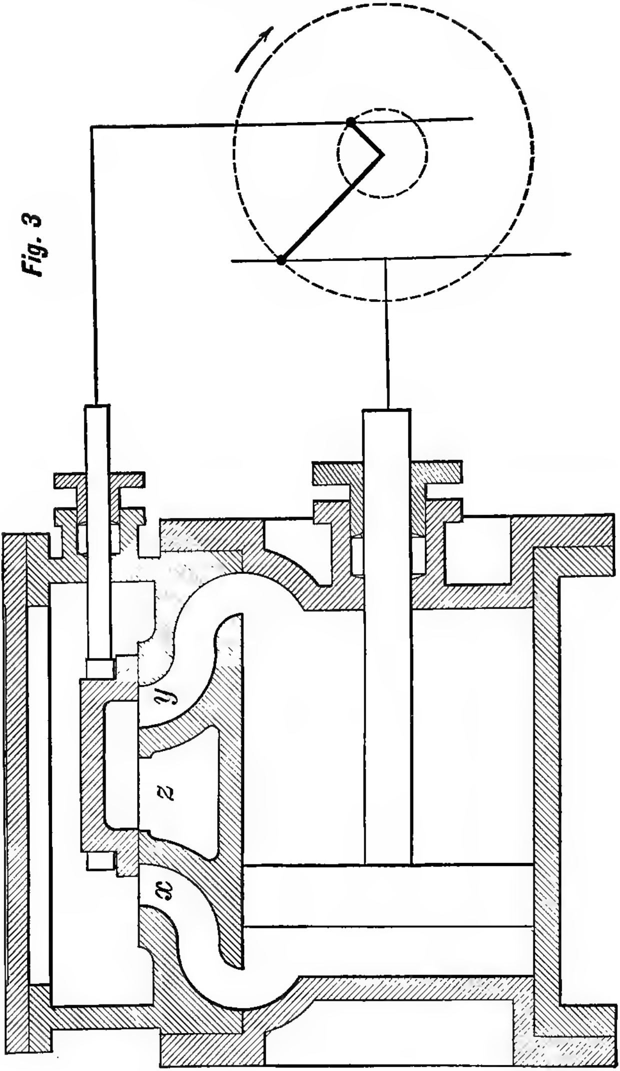
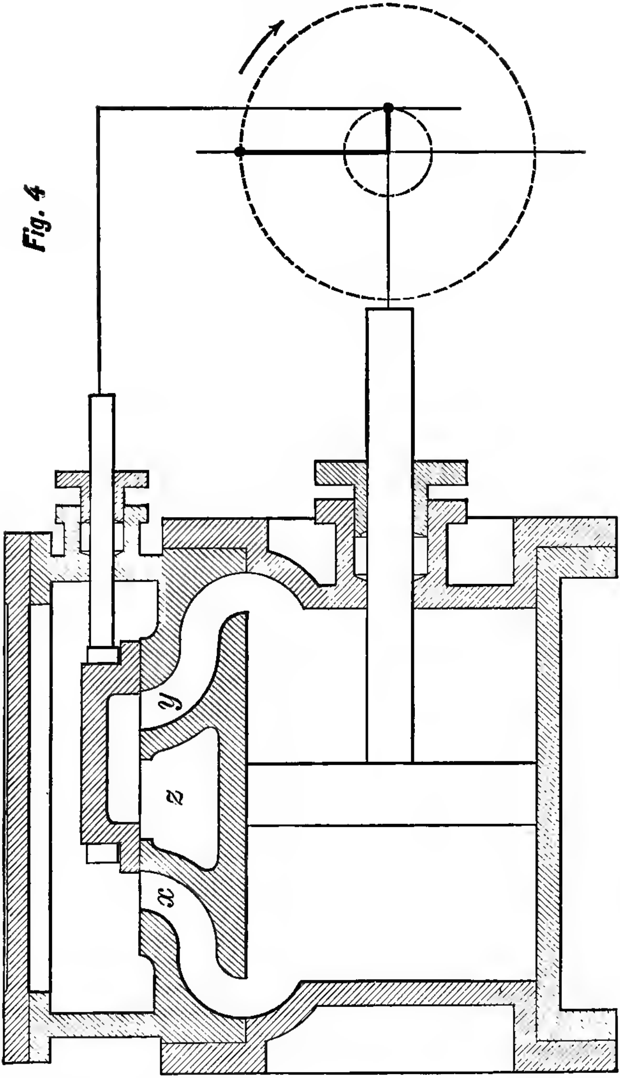


Fig. 3

Fig. 4



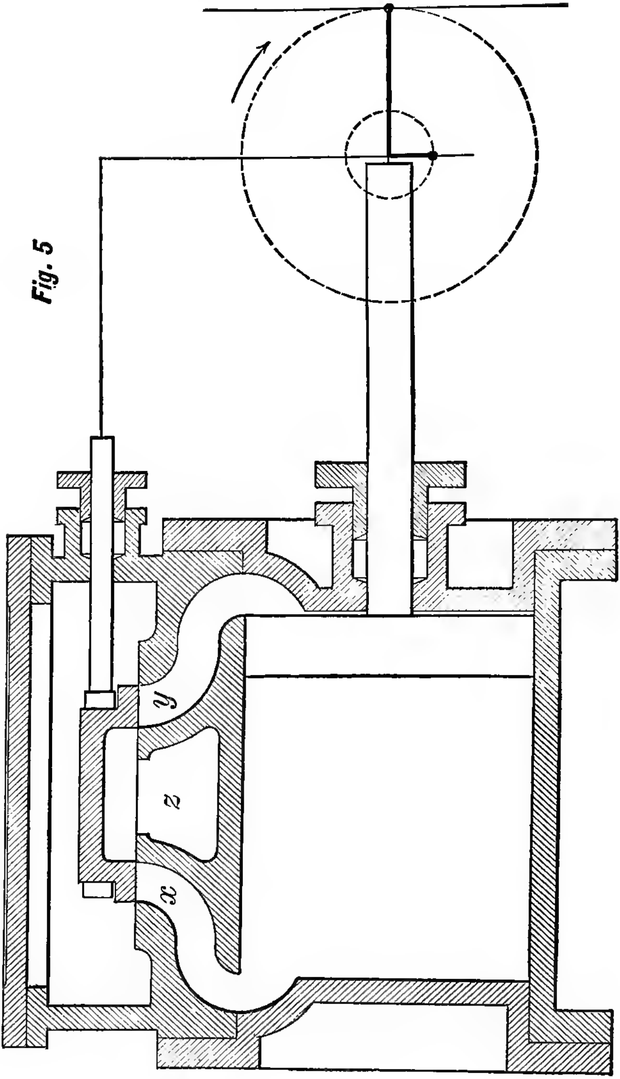


Fig. 5

drive it forward, and opening *y* enabled the steam which previously filled the space in front of the piston to escape to the cavity *z*, which communicates through the exhaust pipe with the atmosphere or condenser, as the case may be. In Fig. 4, the crank shaft has turned through an additional angle of forty five degrees, bringing it at right angles to its initial position. The piston is now at the centre of its travel, and the valve at its extreme right hand position. As the rotation continues, the piston continues to advance; but the valve reverses its motion, and gradually closes its ports, until when the crank completes a half revolution, as shown in Fig. 5, the valve reaches its middle position at which it stood in Fig. 2, with all ports closed. Continuing the motion, the valve is carried to the left, opening port *y* to steam and *x* to exhaust, as shown in Fig. 6, and the piston is driven back to its original position; and this sequence of operations will obviously continue indefinitely.

With the eccentric located as in the figures, the direction of rotation must be as described. This will be apparent if, starting with Fig. 2, rotation in the opposite direction be imagined. The effect of this would be to open port *x* to exhaust and *y* to steam, thereby effectually stopping the rotation in the direction imagined. To effect this reverse rotation the eccentric must be located diametrically opposite to the position shown in the figures.* The student should satisfy himself of

* This is true with the primitive form of valve only. With valves having lap, as actually used, the eccentric position for reverse rotation is *not* diametrically opposite from the position required for forward rotation. This subject will be referred to again.

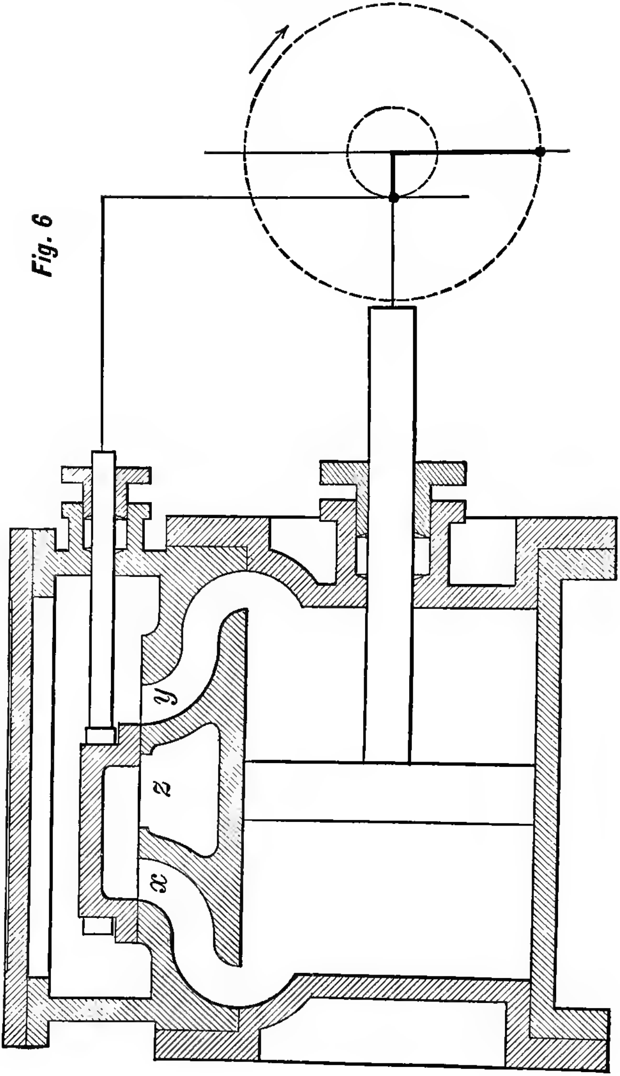


Fig. 6

the correctness of this fact by supposing the eccentric so located, and then following the motion through a revolution.

Throughout this book, whether shown or not, it will be understood that the cylinder is located as in the figures already explained, i.e., to the left of the shaft; and unless otherwise specified, that the direction of rotation is the same as in these figures, i.e., "over."

DEFECTS OF THE PRIMITIVE ENGINE.

With the construction of Figs. 2–6 the opening and closing of the ports are coincident with the passing of the centre by the crank. Economy of steam and successful running require that the following changes be made in this distribution of the steam:

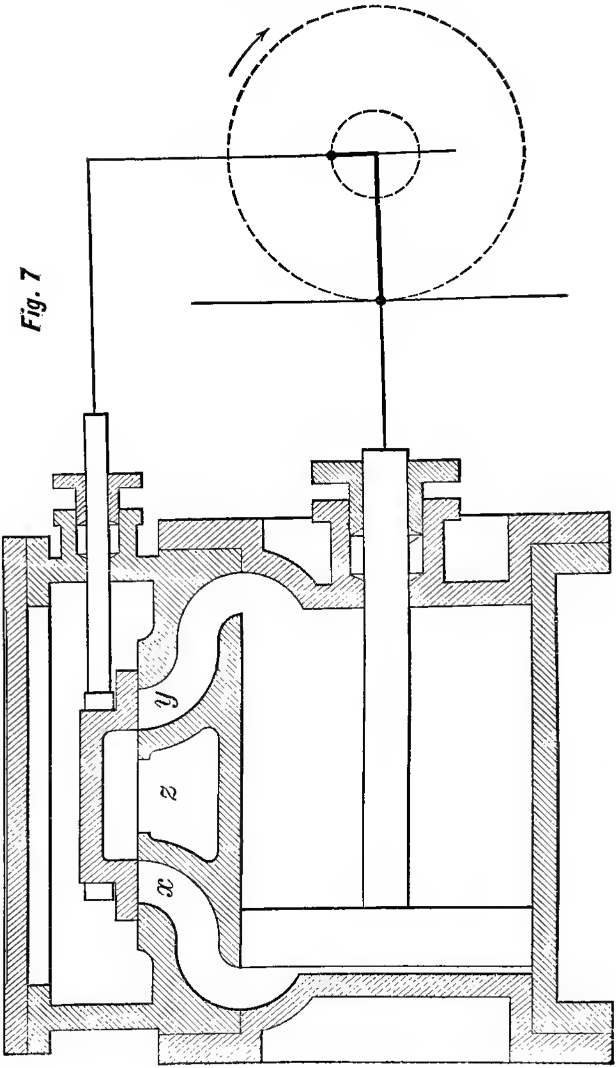
I. The opening of the steam port or "admission" should occur slightly before the crank reaches the centre.* In a general sense this is called giving the valve *steam lead* or simply *lead*. In a more strict sense, that term means the width of opening in fractions of an inch which the valve has given to the steam port at the instant the crank passes the centre.

II. The closing of the steam port or "cut-off" should occur a good deal before the crank passes the centre.

III. The opening of the exhaust port or "release,"

* Of late years a difference in practice has arisen in this respect. Some makers now set their valves to open the port just as the crank passes the centre, and in some cases the admission is delayed until after that event. The statement in the text, however, represents general practice. This subject will be referred to again farther on.

Fig. 7



and its closing or "compression," should occur earlier than the opening of the steam port, but not so early as its closing. As the width of port opening to steam as the crank passes the centre is called steam lead, so the width of opening to the exhaust at the same instant is called exhaust lead or inside lead (compare page 4).

These changes in the steam distribution are brought about by two changes in the valve gear: (I.) The valve is given lap, and (II.) The eccentric is advanced on the shaft ahead of the position given.

LAP.

Fig. 7 is a reproduction of Fig. 2, with the addition of outside or steam lap to the valve. There is no change in the exhaust side of the valve nor in the angular position of the eccentric, and it is obvious that the ports will be opened and closed to the exhaust as the crank passes the centre exactly as before; but the port x will not be opened to steam until the valve has been carried to the right an amount equal to its lap.

This will happen as shown in Fig. 8, when the shaft has turned through an angle *def*, such that *df*, or what is the same thing, *eg*, is equal to the lap. This angle *def* is called the *lap angle*. The valve opens port x to steam when the edge of the valve passes the edge of the port going to the right, and it closes it when the edge of the valve passes the edge of the port going to the left. The position of the valve is the same at closing as at opening, the only difference between the two acts being in the direction of the valve's motion; con-

Fig. 8

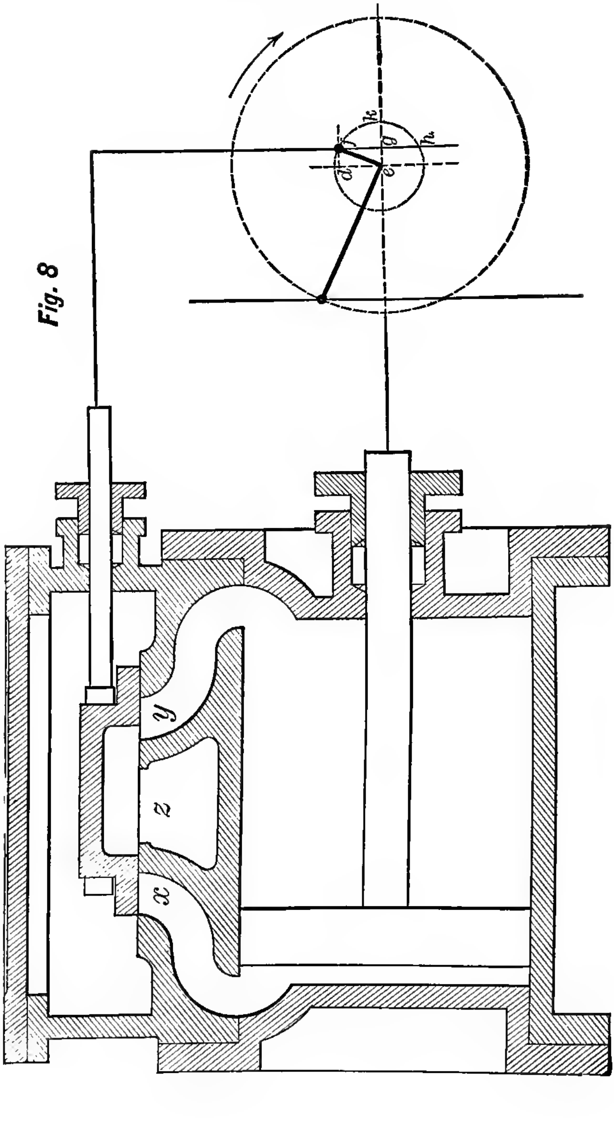
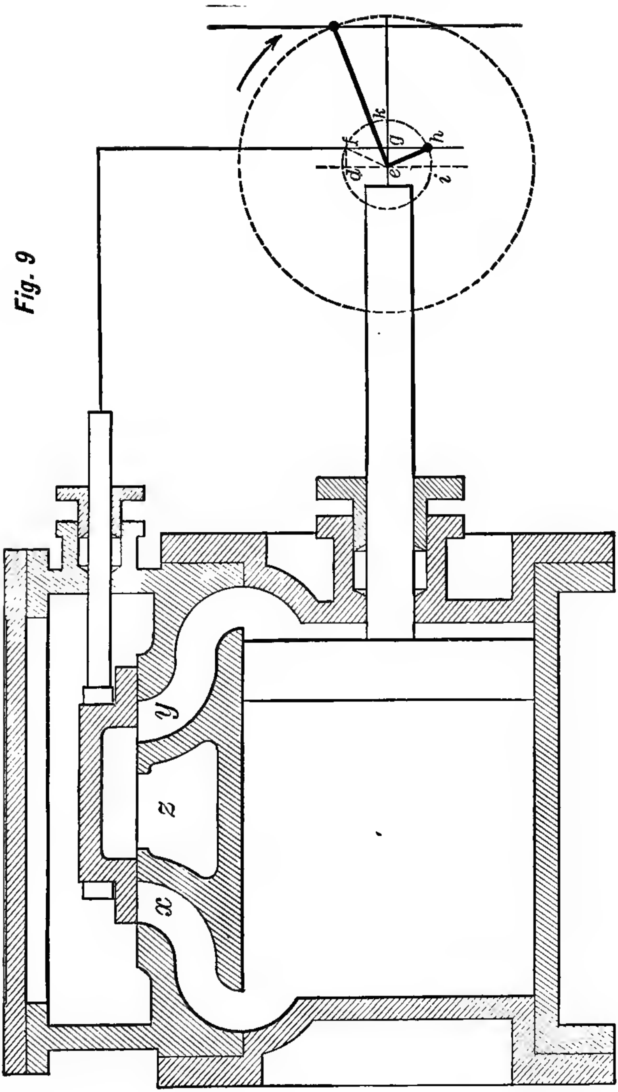


Fig. 9



sequently the port must close with the eccentric at *h*, vertically below *f*, as shown in Fig. 9. From this, two important and fundamental facts can be learned:

I. During the time that steam was being admitted the eccentric (and with it the shaft and crank) turned through the angle *feh*. Now *feh* is equal to a semicircle less *def* and less *hei*, that is, a *semicircle less twice the lap angle*, and this is the first result of the addition of lap to the steam side of the valve. During the admission of steam the crank turns through an angle equal to a *semicircle less twice the lap angle.*

II. In the case of the primitive valve of Figs. 2–6, the port began to open with the eccentric in the position of Fig. 2. It attained its greatest opening in the position of Fig. 4, and the maximum width of port opening was equal to the throw of the eccentric. With the valve of Figs. 7–9, however, the port does not begin to open until the eccentric reaches the point *f*, and the remaining travel *gk* only is available as port opening. This width of opening, *gk*, is equal to *ek* less *eg*, that is, to the *throw of the eccentric less the lap of the valve;* and it follows at once that if a valve with lap is to give the same port opening as another without lap, the eccentric throw of the former must exceed that of the latter, and the greater the lap the greater must be the throw.

ANGULAR ADVANCE.

The last section has shown how, by the addition of lap, the period of port opening may be shortened as desired. While, however, the method of regulating

the length of period of port opening was pointed out, nothing was said about properly timing the opening or closing of a valve having lap with reference to the position of the piston, and in point of fact in Fig. 8 the port opening to steam occurred long after the crank had passed the centre. The correct timing of the events of the stroke, and especially of the admission of steam, is obtained by advancing the eccentric around the shaft from the position thus far shown. In order to give admission to the steam at the instant the crank passes the centre, it is necessary to first locate the crank on the centre, then to advance the eccentric around the shaft such an amount as to draw the valve to the right a distance equal to the steam lap, and finally to secure the eccentric in that position. Such a setting of the eccentric is shown in Fig. 10, in which the eccentric has been turned forward until the distance *df* or its equal, *eg*, is equal to the lap. Such advance of the eccentric on the shaft is called the *angular advance* or the *advance angle* of the eccentric; and if the admission is to occur with the crank on the centre, as in this instance, the advance angle of Fig. 10 is equal to the lap angle of Fig. 8. Having secured a proper admission to the steam, it is proper to inquire next into the effect which this change in the angular position of the eccentric has had on the other events of the stroke. It is sufficiently obvious that turning the eccentric forward a given angle would simply cause each event to occur that much earlier in the rotation of the crank. After steam lap had been added, and before the advance of the eccentric, the cut-off occurred with the crank lacking one lap angle of having reached

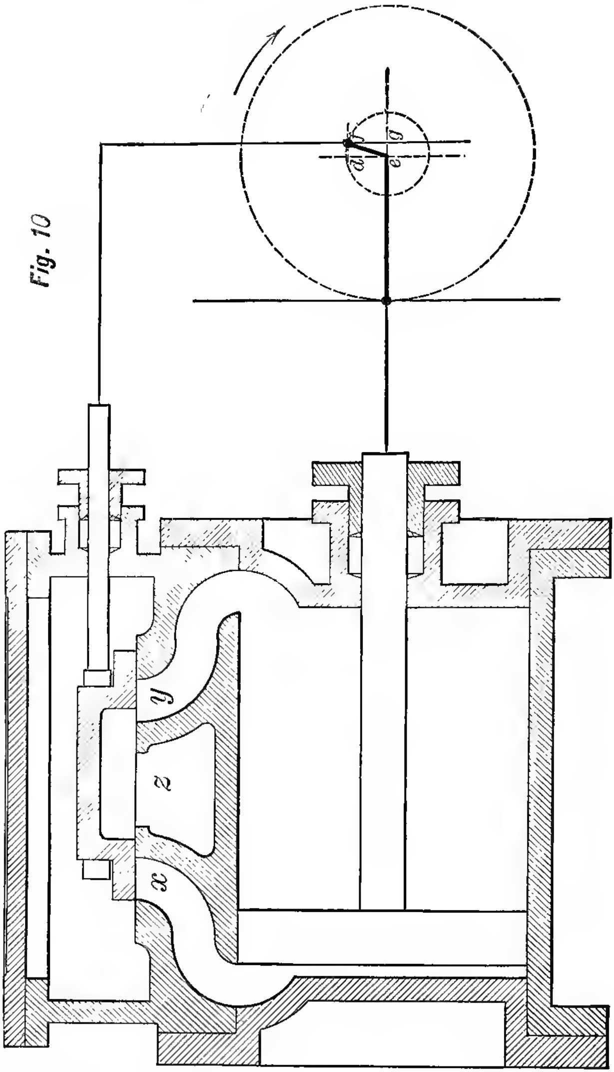


Fig. 10

the centre (see Fig. 9). Advancing the eccentric one lap angle will cause cut-off to occur one lap angle earlier still, or with the crank lacking two lap angles of having reached the centre. Before the advance of the eccentric, the opening and closing of the ports to the exhaust occurred as the crank passed the centre (see Figs. 5 and 7). Advancing the eccentric one lap angle will therefore cause both release and compresssion to occur with the crank lacking one lap angle of having reached the centre.

Summarizing then, the valve has had steam lap added to it, and the eccentric has been advanced by an angle equal to the lap angle, and the resulting steam distribution is as follows: Admission occurs as the crank passes the centre; cut-off occurs two lap angles before the centre; and release and compression occur one lap angle before the centre.

Throughout this book the advance angle will be designated on the diagrams by the letter δ (delta).

LEAD.

It was explained on page 13 that the admission of steam should take place slightly before the crank reached the centre, and that such early admission was called lead. In the last section, for the sake of simplicity, the admission was supposed to occur as the crank passed the centre. In other words, the lead was made zero. It now becomes in order to examine the method for the introduction of lead, and the changes in the other events of the stroke which follow. In Fig.

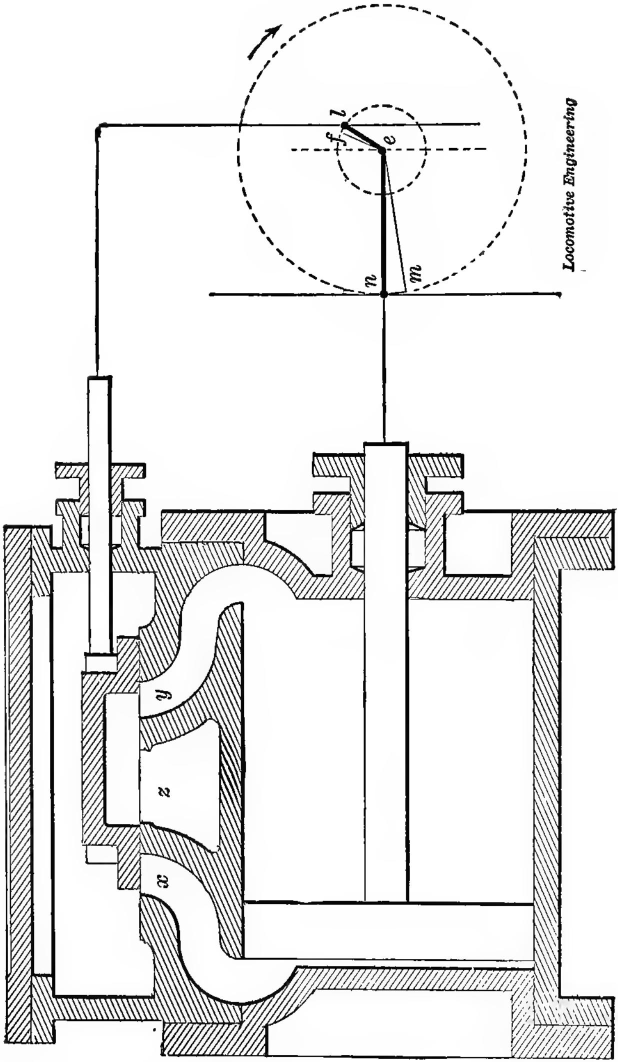


Fig. 11

10 the eccentric was advanced to cause admission to occur on the centre. If it is proposed to give the valve lead, the eccentric must be advanced still further, so as to draw the valve to the right an additional amount equal to the lead desired. In Fig. 11 this additional advance has been made, the eccentric having been moved from *f* of Fig. 10 (reproduced in Fig. 11) to *l*. The angle *fel* is called the *lead angle*, and it follows that in all cases the angular advance is equal to the lap angle plus the lead angle. If the lead is zero, then, as before found, the angular advance is equal to the lap angle. If in Fig. 11 the crank shaft be turned backward, the valve will close the port when the eccentric reaches point *f* and the crank stands at *m*, the angle *nem* being equal to the angle *fel*. In other words, the lead angle is equal to the angular distance which the crank lacks of having reached the centre when admission occurs.

It was found on page 19 that advancing the eccentric on the shaft a given angle advanced all the events of the stroke correspondingly, and the resulting distribution of steam with no lead was summarized on page 21. If now the valve have lead so that the angular advance be greater than the lap angle, the steam distribution given on page 21 is changed as follows:

Admission occurs with the crank lacking the lead angle of having reached the centre.

Cut-off occurs with the crank lacking two lap angles and one lead angle of having reached the centre.

Release and compression occur with the crank lacking one lap angle and one lead angle of having reached the centre.

An application of the above principles is all that is

necessary for the analysis of the steam side of any existing plain slide valve, as will be seen from the following example:

An eccentric has a throw of $1\frac{5}{8}''$, the valve has one-

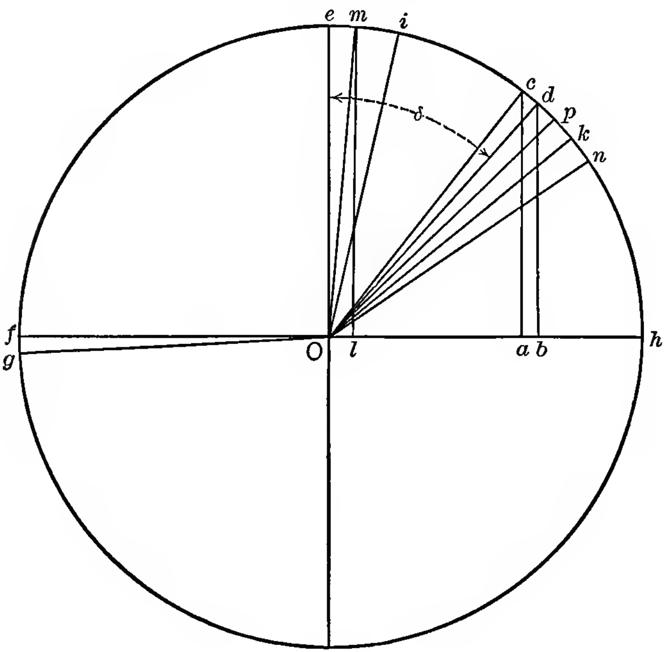


Fig. 12

inch steam lap, no exhaust lap, and the eccentric is set to give $\frac{1}{16}''$ lead. Required the greatest port opening, and the crank positions for admission, cut-off, release, and compression.

In Fig. 12 strike the circle with a radius equal to the

throw of the eccentric, and lay off *Oa* equal to the lap of the valve and *a b* equal to the lead. Erect perpendiculars from points *a*, *b*, giving points *c*, *d*. Now *eOc* is the lap angle, *cOd* is the lead angle, and *eOd* is the angular advance. Lay off *fg* equal to *cd*, giving *Og* the crank position for admission. Make *hi* equal to twice *ec* plus *cd*, giving the crank position *Oi* for cut-off. Make *kh* equal to *ec* plus *cd*, giving *Ok* the crank position for release and compression. Finally, *ah* is the greatest steam port opening.

The student should familiarize himself with the principles thus far treated, by the solution of a variety of problems similar to the following:

Problem I. Eccentric throw $1\frac{3}{8}$, lap $\frac{3}{4}$, lead $\frac{1}{16}$. Required crank positions for lead, cut-off, compression and release, and port opening to steam.

Problem II. Eccentric throw $1\frac{3}{4}$, lap $\frac{13}{16}$, lead $\frac{3}{32}$. Required as before.

EXHAUST LAP.

The action of exhaust lap upon release and compression is precisely the same as that of steam lap upon admission and cut-off. The proportions existing between exhaust lap and angular advance are, however, quite different from those between steam lap and angular advance—a fact which in a measure obscures the really identical action of lap on the two edges of the valve, and renders some special attention to exhaust lap desirable. The sum of the steam lap and lead angles

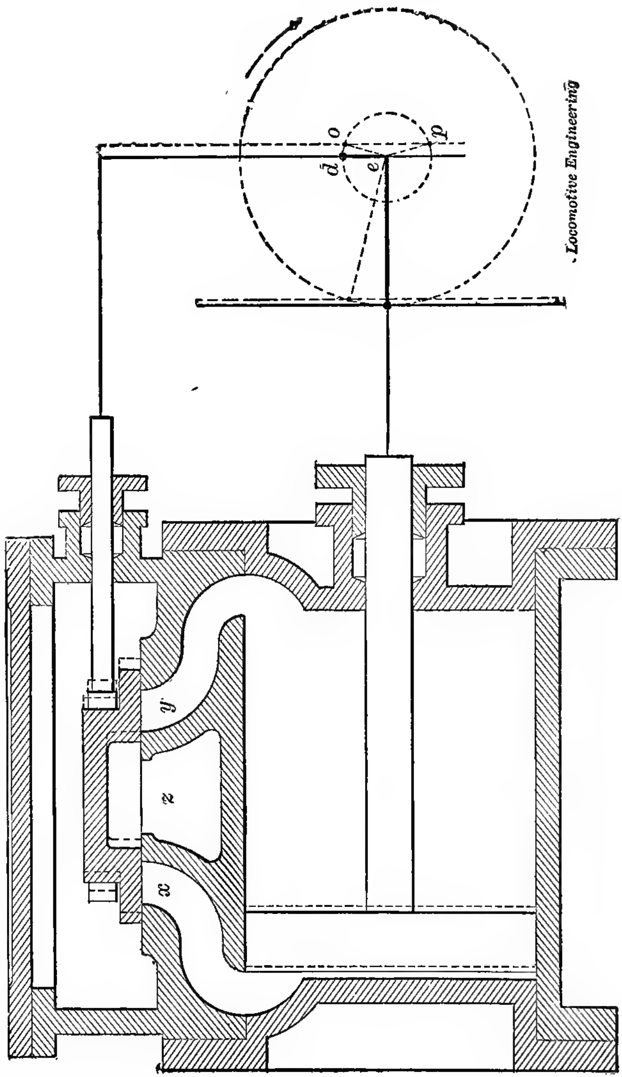


Fig. 12a

has already been shown to be equal to the angle of advance, and in precisely the same way the sum of the exhaust lap and lead angles is also equal to the advance angle. Considering the steam side, however, the lead angle is but a small part of the advance angle—the lap angle being nearly equal to the advance angle. The exhaust lap is always much smaller than the steam lap, the exhaust lap angle is correspondingly small and the exhaust lead angle correspondingly large—being in fact the larger of the two.

Fig. 12*a* shows in full lines the parts in the same position as in Fig. 7, with no angular advance to the eccentric, but with the addition of inside lap to the valve. It will be apparent that port *y* will be opened to exhaust when the valve has moved to the right an amount equal to the exhaust lap, which will happen when the shaft and eccentric have turned to the position shown by the dotted lines, such that the horizontal distance *do* is equal to the exhaust lap, and the port will close again when the eccentric has turned through the angle *oep* to the point *p* vertically below *o*. The angle *deo* is obviously the exhaust lap angle, and the angle *oep*, during which the port remains open to the exhaust, is equal to a semicircle less twice the exhaust lap angle, precisely as with the steam side of the valve.

In Fig. 12*b* the eccentric is shown turned forward on the shaft to give steam lead as in Fig. 11, point *o* being reproduced from Fig. 12*a*. It is evident that turning the eccentric from *o* to *l* drew the valve to the right and gave exhaust lead, the angle *oel* being the exhaust lead angle, as *fel* of Fig. 11 is the steam lead

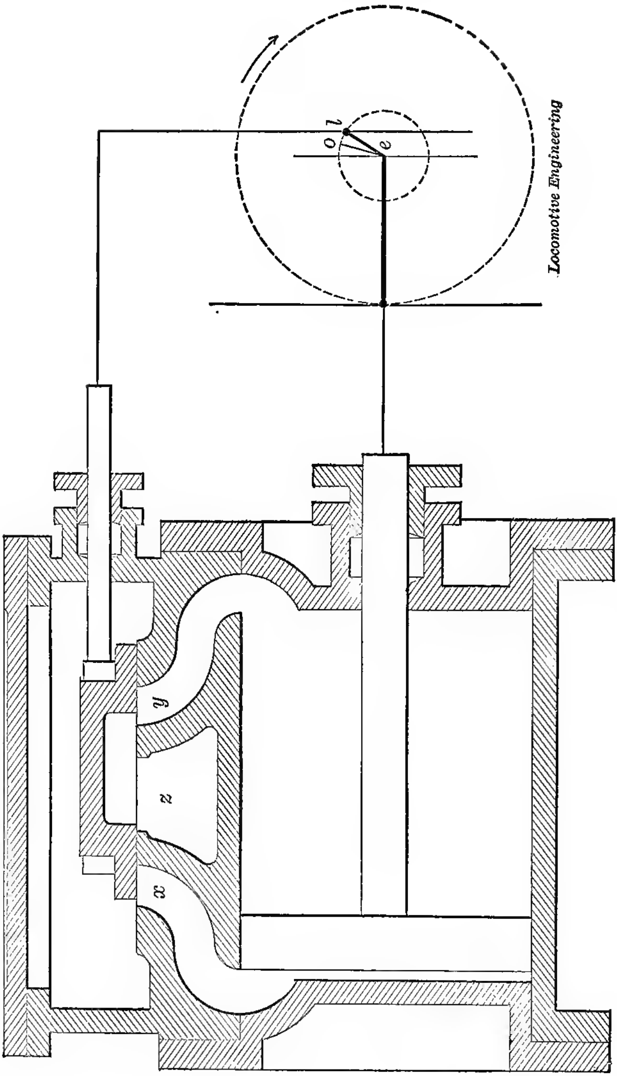


Fig. 12b

angle. It will thus be seen that the action of exhaust lap does not differ from that of steam lap, except in degree, and a moment's reflection will show that the extreme opening of the port to exhaust is equal to the throw of the eccentric less the exhaust lap precisely analogous to the steam port opening as stated on page 18.

Furthermore, companion statements relating to exhaust opening and closure could be made to those on page 23, relating to steam lead and cut-off, but these points for a valve having exhaust lap are more usually stated with reference to what they would be if the valve had no such lap.

Reference to Fig. 12*a* will show that the effect of exhaust lap is to delay the opening and hasten the closing of the port, and in each case by an angle of rotation of the shaft equal to an exhaust lap angle; that is, release occurs a lap angle later and compression a lap angle earlier than they would if the valve had no exhaust lap.

As stated on page 4, exhaust lap is frequently absent and even negative (in which latter case it is often, though without much significance, called inside clearance). The effects of negative exhaust lap are of course the opposite of those of positive lap. That is, negative lap hastens the opening and delays the closing of the port, and, as with positive lap, in each case by an angle of rotation of the crank equal to the (negative) lap angle, and while with positive lap the period or port opening is equal to a semicircle less two lap angles, with negative lap it is equal to a semicircle plus two lap angles.

Negative inside lap also acts to increase the maximum port opening beyond the amount due to a valve made "line and line," the opening being equal to the throw of the eccentric plus the numerical value of the negative lap.

BACKWARD ROTATION.

It was explained on page 11 that with a primitive valve the eccentric location for rotation in the reverse direction would be diametrically opposite that shown in Figs. 2–6. For a valve having lap, the position for reverse rotation is found by laying off the advance angle in the direction of the proposed rotation from the position for a primitive valve. The effect of a rock shaft in the valve motion (for an example of which see any American locomotive) is to reverse the motion of the valve as compared with the eccentric, and hence to require a location of the eccentric which will provide for this reversal. The position of the eccentric for either direction of rotation, and with or without a rocker, may be located from the following facts:

I. Without a rocker the eccentric for a primitive valve is 90° in advance of the crank in the direction of the rotation. With a rocker the eccentric is behind the crank.

II. The advance angle is laid off in all cases in the direction of the rotation from the position for a primitive valve.

One qualification should be added to the above, as follows: In all the cases thus far shown, the location of

the eccentric for the primitive valve is as stated at right angles to the crank. In certain cases, owing to the character of the connections between the eccentric and valve, this is not true. For an example see Figs. 58 and 59. In cases of this kind the location of the eccentric can be found as follows: Carry the centre of the eccentric strap to the centre of the crank shaft. Through the centre of the shaft draw a line perpendicular to the location of the eccentric rod thus found. This perpendicular gives the location of the eccentric for the primitive valve, and the angular advance is to be laid off from it in the direction of the rotation.

THE BILGRAM DIAGRAM.

Any existing slide valve can be analyzed by the methods that have been followed in explaining the action of the valve, and new valves could be designed by a tentative application of the same methods. Such a plan of procedure, however, would be exceedingly tedious, and much ingenuity has been expended in devising briefer and better methods. Of these, by far the best is the diagram devised by Mr. Hugo Bilgram, and explained below. The chief office of such a diagram is to show briefly and accurately the position of the valve for any and every position of the crank.

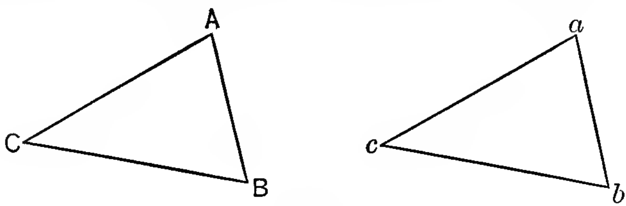


Fig. 13

The demonstration of the Bilgram diagram depends upon the following theorem of geometry: In Fig. 13 let ABC and abc be two triangles, such that any two of their angles, as those at A, C and a, c, and any one side, as BC and bc, are respectively equal. Then this theorem asserts that all of the other parts of the triangles are equal, i.e., angle B to b, side AC to ac, and side AB to ab.

In Fig. 14 let A be the dead-point location of the crank, and B be the corresponding position of the ec-

centric centre, δ being the angle of advance. It is obvious enough that the valve is now located a distance Bb (equal to the sum of the lap and lead) to the right of its middle position. Imagine the crank to turn through the angle α to a new position A'. The eccentric will turn through an equal angle α to its new

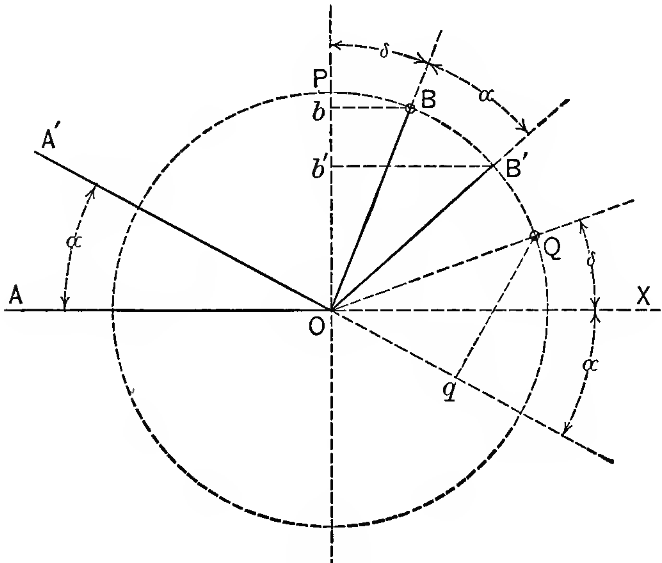


Fig. 14

position B', and the valve will then be located a distance $B'b'$ to the right of its middle position. Lay off the angle δ upward from OX, and thus locate a fixed point, Q. From Q drop Qq perpendicular to the new crank position extended. There are thus formed two triangles, $B'b'O$ and QqO, and in them $B'O$ equals

QO, since both are radii of the same circle. Angles $B'b'O$, QqO are equal, because both are right angles; and finally, angle $B'Ob'$ equals QOq, since each is equal to δ plus α. The two triangles have thus two angles and a side of one, respectively equal to two angles and a side of the other, and it follows that the triangles are equal in all their parts, and hence Qq equals $B'b'$. $B'b'$ is the distance which the valve has travelled from its central position for crank position A', and it hence follows that Qq likewise equals that distance. The same demonstration can be made for any other crank position as well as for A', and the following general fact is thus established: Lay off the advance angle above the centre line, and thus locate the fixed point Q. Draw any crank position desired, and extend it if necessary. From the fixed point Q drop a perpendicular to the crank line, and the length of the perpendicular will be equal to the distance of the valve from its central position for the crank position taken. For rotation in the reverse direction, points B and Q would fall below instead of above the centre line.

The length of the perpendicular Qq gives the distance of the valve from its central position, but it does not of itself show whether the valve is located to the right or to the left of its middle position. That fact will be determined instinctively after a little practice in the use of the diagram; but if desired it can be determined by the following consideration: Referring to Fig. 14, that side of the crank and of its imaginary extension facing the space toward which the crank is revolving may be called the face side of the crank, and the opposite side may be called the rear side. If the

perpendicular *Qq* falls upon the face side of the crank, the valve is to the right of its middle position; if the perpendicular falls upon the rear side, the valve is to the left of its middle position.* As has been explained, the greatest port opening is equal to the throw of the eccentric *OQ*, Fig. 14, less the lap; and it is also true that for any position of the crank, the port opening which exists at that position is equal to the displacement of the valve from its central position less the lap, i.e., to the value of *Qq* for that position, less the lap. In other words, if for any crank position the value of *Qq* be found, and from it the steam lap be taken, the result will be the distance which the steam port stands open for that crank position. If, on the other hand, the exhaust lap be taken from it, the result will be the distance which the exhaust port stands open. This subtraction can be conveniently made by striking two circles *L*, *l* from *Q* as a centre, and with radii equal to the steam and exhaust laps respectively, as is done in Fig. 15. In case the inside lap is negative, it of course increases instead of decreases the port opening to exhaust. Throughout this book, positive lap will be shown by full circles, and negative lap by dotted circles.

Starting with the position *A*, Fig. 15, the length of the perpendicular which locates the valve is *Qq*, and the width of opening of the port to steam is *aq*; *A* being the dead-point position of the crank, *aq* is the lead of the valve. Similarly, *bq* is the exhaust lead. In Fig. 16 the valve is shown in position for crank position *A*.

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* This takes it for granted, as explained on page 13, that the position of the cylinder is to the left of the shaft.

The opening of the port *c* to steam is equal to *aq* of Fig. 15, and the opening of port *d* to exhaust is equal to *bq*. Similarly, the displacement *e* of the valve from its centre is equal to *Qq*.* As the crank revolves, *Qq* gradually lengthens until the crank reaches position *B* perpendicular to *OQ*, when *Qq* becomes *QO*, which is its

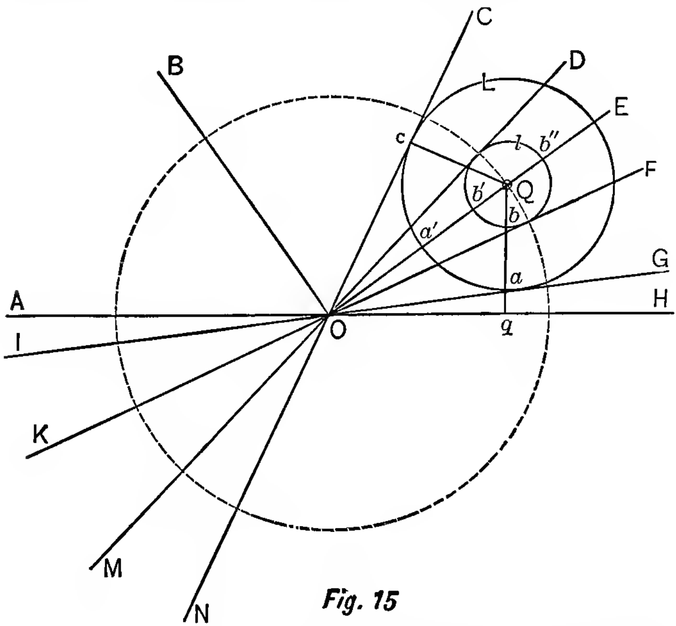

Fig. 15

greatest value. The valve now stands at its extreme right hand position as shown in Fig. 17, the ports being open to their greatest amount—the steam port by a

* It will be understood that the distances stated as equal are not so shown in the cuts, as they are necessarily drawn to different scales.

width $a'O$, and the exhaust port $b'O$. Passing B, the valve returns towards its central position, and at C the

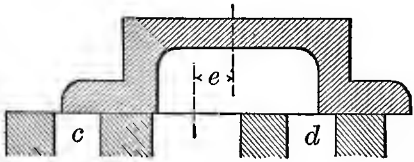

Fig. 16

displacement has been reduced to equality with the steam lap. The port c is therefore closed to steam, and

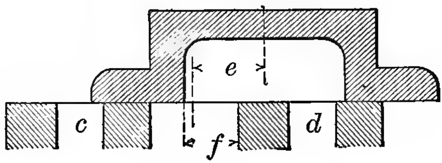

Fig. 17

cut-off takes place as shown in Fig. 18. At D, port d is closed to the exhaust, Fig. 19, and compression be-

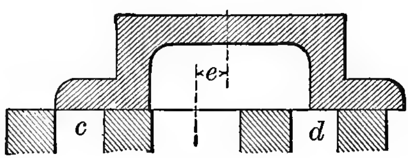

Fig. 18

gins. At F, port c is opened to exhaust, Fig. 20, and release occurs. At G the valve is ready to open port d for the return stroke, Fig. 21 ; and at H the valve has

opened port *d* by the amount of the lead. The positions of the crank for the return stroke are readily found

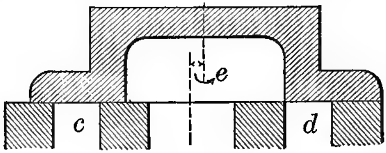

Fig. 19

by extending the crank lines beyond the centre. Thus *I* is the lead position, *K* the release, *M* the compression,

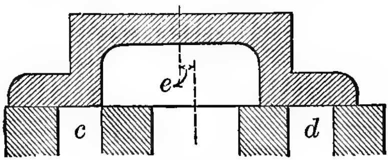

Fig. 20

and *N* the cut-off. Had there been no exhaust lap, release and compression would have occurred simultaneously at *E*; and had the exhaust lap been negative, release and compression would have exchanged places, and the maximum opening to exhaust would have been *Ob''*.

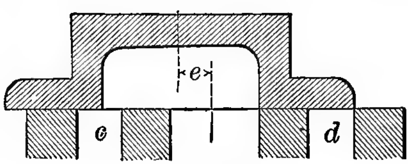

Fig. 21

Consideration of this diagram will recall and enforce the essential effect of lap, as stated on page 18 ; i.e., to shorten the period during which the port is open. Thus with steam lap the steam port is open when the crank is moving from I to C and from G to N, and with exhaust lap the exhaust port is open from K to D and from F to M. With negative inside lap, on the other hand, the port is open during more than half a revolution, i.e., from M to F.

The principles laid down should be fixed in the mind by the solution of practical problems similar to the following:

Problem III. Throw of eccentric 2″, steam lap $1\frac{1}{8}$, exhaust lap $\frac{1}{4}$, lead $\frac{1}{8}$. Required port opening and points of cut-off, release, and compression.

Problem IV. Travel of valve 3″, steam lap $\frac{1}{2}$, negative exhaust lap $\frac{3}{16}$, lead $\frac{1}{8}$. Required as in the last problem.

This diagram is of use not only in analyzing existing valve motions as in the preceding problems, but also in designing new ones to meet required conditions. The method of using it for this purpose is best shown by an illustrative example, as follows:

The valve for a certain engine is to have a steam-port opening of $\frac{5}{8}$″, a lead of $\frac{1}{16}$; is to cut off the steam at $\frac{3}{4}$ of the stroke, and open the exhaust at 95 per cent of the stroke. Required the inside and outside lap, the throw and advance angle of the eccentric, and the point of exhaust closure.

In Fig. 22 make AB equal to the length of the stroke, using a scale of three inches to the foot. Make Aa

equal $\frac{3}{4}$ of AB, and Ab equal .95 of AB. Draw the semicircle $A'a'b'B'$ to represent the path of the crank, and project to it the points a, b. Draw Oa' and Ob', which are the crank positions for cut-off and release. Draw cd such that de is equal to the lead opening, $\frac{1}{16}$

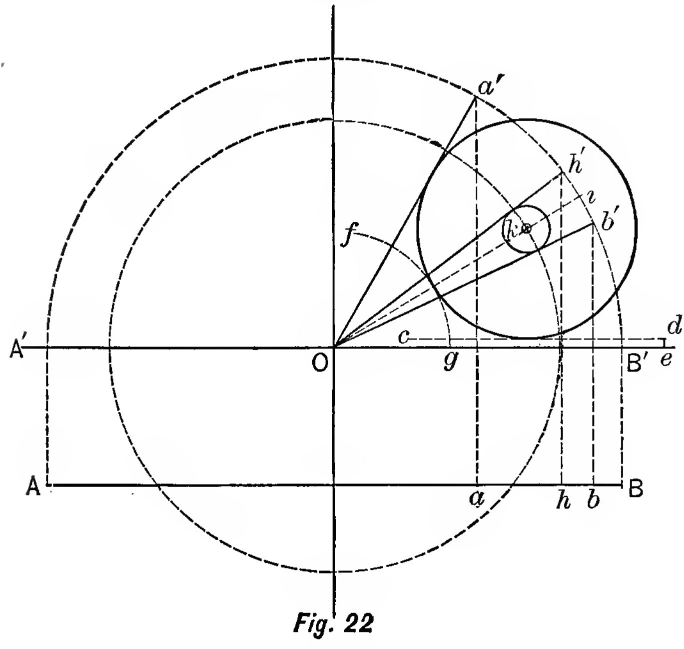

Fig. 22

inch; and strike the arc fg with radius equal to the port opening. Find by trial the centre and radius of the steam lap circle such that it shall be tangent to Oa', cd, and fg. From the same centre strike the exhaust lap circle tangent to Ob'. Draw Oh' tangent to the exhaust lap circle and project h' to the stroke line, giving h.

Measuring the diagram, the results sought are, outside lap $\frac{9}{16}$ inch, inside lap $\frac{1}{8}$ inch, throw of eccentric $1\frac{3}{16}$ inch, advance-angle iOB', point of exhaust closure h, which is 90 per cent of the stroke. It may be observed further, that the exhaust port opening is Ok; in this

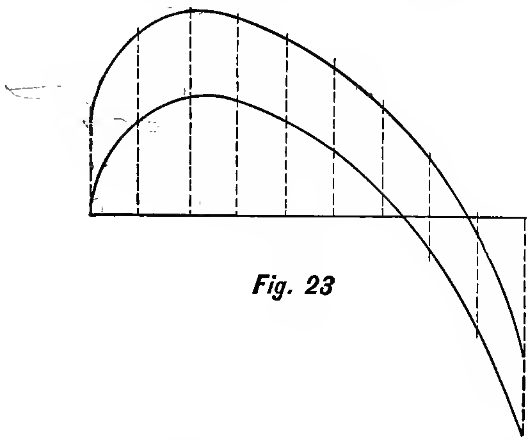

Fig. 23

case and in all others this is more than sufficient for the purpose, and hence no particular care is necessary in relation to it.

An interesting and profitable exercise in this connection is to make a diagram showing by a continuous line the varying width of port opening throughout the stroke. This is illustrated in Fig. 23, in which the continuous base line represents the stroke of the piston of Fig. 22 divided into tenths. At each division a perpendicular is erected, and on this perpendicular is laid off the opening of the ports to steam and exhaust for that position of the piston—this opening being obtained from Fig. 22. Through the points thus found the

curved lines are drawn—the upper one for the exhaust port and the lower one for the steam port. The crossing of the base line by the curved lines shows the points of cutting off and compression, respectively; and the extension of the curved lines below the base line shows the distances by which the edges of the valve have closed the ports. This diagram shows at a glance how gradual is the cutting off of the steam. Such diagrams are exceedingly useful in connection with the study of independent cut-off valves, many of which will not give flattering results when subjected to this analysis.

LAYING OUT THE SLIDE VALVE.

The diagram Fig. 22 gives all the dimensions necessary for laying out its valve, except the width of the exhaust cavity, and that is determined at once by drawing the valve and its seat with the valve at one extreme of its travel, that is, in the position already shown in Fig. 17. Referring to Fig. 1, it is clear that the width of the acting face of the valve is equal to the outside lap *plus* the width of the port *plus* the inside lap. In order to determine all the dimensions of the valve face and seat proceed as follows: Lay down the width of the left hand port (rules for which will be given farther on) and the width of the bridge (usually made equal to the thickness of the cylinder). On these locate the acting face of the valve with the port open to steam to the greatest amount intended. From the exhaust edge of the acting face lay off the distance *f*, Fig. 17, to equal or slightly exceed the width of the port, thus completely determining the exhaust cavity in the cylinder.

From the right hand edge of this cavity the remaining bridge and port are to be laid off the same as the left hand side. The valve seat being completed, and the steam and exhaust laps being known, it is easy to complete the drawing of the valve. The proper determination of the distance *f*, Fig. 17, as above, is all that need be considered in designing the exhaust cavity in the cylinder. If this cavity be made too narrow, it will cramp the exhaust; if too wide, it will add unnecessarily to the size of the valve and to the steam pressure upon it, and hence to the friction and wear and tear on all the valve gear. Further than this, the size of the exhaust cavity has no influence on the valve motion.

It will be observed that in the valve diagram Fig. 22, the lines for the piston and crank circle are drawn to a reduced scale, but the lines for the valve and eccentric are full size. The reduced scale for the crank dimensions is for convenience. The valve dimensions should always be made full size.

* VELOCITY OF THE VALVE.

Referring to Fig. 14, it is obvious that the valve will move with its greatest velocity when the eccentric is at *P*. At this point its velocity may be represented by the eccentric throw *OP*. At any other position of the eccentric as *B*, the valve will move with a velocity proportional to the leverage with which the eccentric acts upon it, that is, *Ob*. Similarly at *B′* the velocity of the valve will be represented by *Ob′*. In the original demonstration of this diagram it was shown that the triangles *B′b′O* and *QqO* are equal in all their parts. It

hence follows that Oq equals Ob'. In other words, if a perpendicular be drawn from the point Q to any crank line, the distance from the centre of the shaft to the foot of that perpendicular will represent the velocity with which the valve is moving with the crank in that position. Quick closure of the port in cutting off steam is considered a merit in a valve motion, and this property of the Bilgram diagram furnishes a ready means of comparing the merits of different valve gears in this respect. In Fig. 15 the perpendicular Qc will determine the distance Oc, which represents the velocity with which the valve is moving at the instant of cutting off.

LIMITATIONS OF THE PLAIN SLIDE VALVE.

Careful study of the Bilgram diagram will explain the features of the common slide valve which have usually been considered to limit its application to cases where a comparatively late cut-off was to be employed. Thus, in the example of Fig. 22, let the given conditions be the same, except that cut-off is to be at half stroke instead of three quarters, and let there be no inside lap. The results are shown in Fig. 24, where it will be seen that the throw has increased to two inches and the steam lap to one and three eighths inches, while the common point of compression and release has gone back to bh—85 per cent of the stroke. At still earlier points of cut-off these features become still more marked, the travel of the valve rapidly increasing and the release and compression becoming more and more premature. The increased lap and travel increase directly the size and duty which the

parts have to perform. Further, as will be seen by referring to the section on laying out the slide valve, and to Fig. 17, they increase the size of the exhaust cavity, and so add to the size of the valve and to the steam pressure upon it. The release can be made later

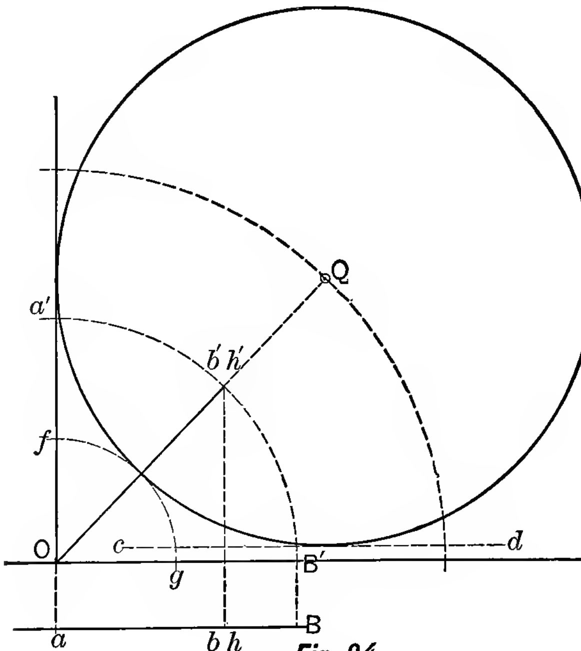

Fig. 24

by the addition of exhaust lap, but this involves a still earlier compression. From these considerations it has been generally held and taught that the plain slide valve could not be profitably employed for cut-offs shorter than one half or five eighths stroke. The

methods which have been adopted to overcome the above difficulties will form the subject of a later chapter,

THE AREAS OF THE PORTS AND PIPES.

It will be seen from the foregoing that the width of port opening is an essential factor in the design of a valve motion. The exact meaning of the term port opening in this connection should be clearly understood. By that term is to be understood the extreme distance of the steam edge of the valve from the steam edge of the port. This distance may, and often does, exceed the width of the port—that is, the valve may have over-travel to secure certain real or fancied advantages. In engines with fixed eccentric, which are now under consideration, the only benefit of such over-travel is to increase the sharpness of the cutting off. This, in the author's opinion, is not worth its cost, and hence he does not practise nor recommend it. In locomotives and shifting eccentric engines the travel of the valve is shortened at the early cut offs, and in such engines, in order to secure sufficient port opening at the early cut-offs, it is proper and necessary to give over-travel at the late ones.

It is clear that the area of the port opening should have a proper relation to the size and speed of the engine. It will be furthermore clear without extended explanation, that in engines having separate admission and exhaust ports (for example, the Corliss) the exhaust passage should have a greater area than the admission passage. In engines using the same passage for both purposes, to which this book relates, that passage should be proportioned to meet the requirements of

the exhaust, and then, if desired, it need not be opened to steam any wider than is necessary for its use as a steam port.

In determining the area of a pipe or passage it is treated as though the velocity of the steam through it were equal to the velocity of the piston multiplied by the ratio of the area of the piston to the area of the port or pipe. Of course, owing to the elasticity of the steam, the varying velocity of the piston and the fact that the port is not constantly opened to its full width, this is not true. The rules should however be regarded as purely comparative between engines in which these factors have substantially the same values and hence may be ignored.

As the result of experience and experiments, the proper velocities of the steam through the various passageways on the above basis are as follows:

Through the steam pipe 8000 feet per minute.

Through the exhaust port 6000 feet per minute.

Through the exhaust pipe 4000 feet per minute.

For free admission of steam the port should be opened three fourths of its width. From the above data the following table is constructed for convenient use:

Piston Speed, Feet per Minute.	Diameter of Steam Pipe (Diameter of Piston = 1).	Diameter of Exhaust Pipe (Diameter of Piston = 1).	Area of Exhaust Passage (Area of Piston = 1).
200	.158	.223	.033
250	.176	.248	.042
300	.194	.272	.050
350	.209	.294	.058
400	.224	.314	.067
450	.237	.333	.075
500	.250	.353	.083
550	.260	.368	.092
600	.274	.385	.100

The table determines the diameters of the steam and exhaust pipes at once, but it gives the area only of the port, leaving its length and breadth to be determined by the designer. The practice in this particular is very diverse. In shifting eccentric automatic engines, which form the subject of Part II, and in which every expedient must be employed to secure sufficient port opening, the length of the ports is often made to equal or even exceed the diameter of the cylinder; but in plain slide valve engines of the usual type, a length of about three quarters the cylinder diameter more nearly represents average practice. This length determined, it is only necessary to divide the area of the passage by it to determine the width of the port, and three fourths of this will give the port opening to be used in laying out the diagram.

A " rule of thumb" which is in very common use is to make the steam pipe one fourth the diameter of the cylinder, and the exhaust pipe one third. At slow speeds this rule gives an excess of capacity over the requirements, to which of course there is no objection; but at high speeds it gives a deficiency. On high grade engines, where the best results are sought, steam pipes are seen as large as one third and exhaust pipes one half the cylinder diameter.

All the principles thus far given will be found required in the solution of the following

Problem V.—An engine with a 10″ × 15″ cylinder is to run at 200 revolutions per minute. Cut-off is to be at $\frac{5}{8}$ stroke, release at .93 stroke, and lead is to be $\frac{3}{32}$″. Required the diameters of steam and exhaust pipes, the

dimensions of the ports, the travel, and the steam and exhaust laps of the valve.

* THE ANGULAR VIBRATION OF THE CONNECTING ROD.

As has been explained, the slotted cross-head was adopted in the preceding to avoid certain distortions which are incident to the use of the connecting rod. It is now proper to discuss these distortions, and explain the methods for neutralizing their effects.

With the slotted cross-head the position of the piston

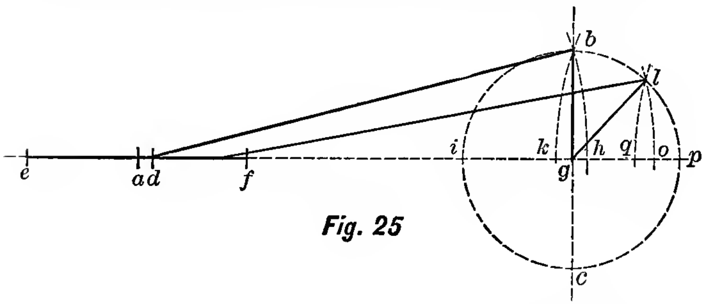

Fig. 25

or cross-head in its stroke for any crank position is found by simply projecting the crank position to its horizontal diameter, or a line parallel thereto, by means of a straight projecting line, as was done in Figs. 12, 22, and 24. Fig. 25 is a skeleton diagram of the usual connecting rod and crank. It is obvious that if the crank pin end of the connecting rod be disconnected from the crank pin and carried to the centre of the crank shaft the cross-head pin will occupy its central position *a*. If from this position the crank pin end be carried to either "quarter" position of the crank pin *b*, *c*, the cross-head

pin will be drawn toward the shaft and will occupy the position *d*. For the forward stroke the position of the cross-head is measured from *e* as a starting point, and hence the cross-head and piston have moved *too far* by the distance *ad*. For the return stroke the position is measured from *f* as a starting point, and hence the cross-head and piston have *not moved far enough* by the same distance. If the valve motion were laid out by the preceding methods to cut off steam at half stroke, it would in fact cut off later than half stroke for the forward stroke and earlier for the return. The same distortion takes place at all other positions of the crank except at the centres, though to a less degree; and it follows that all the events of the stroke except the lead occur too late in the forward stroke and too early in the return. The amount of this distortion can be found for any position of the parts, as is done in Fig. 25, for the position shown, by striking an arc with radius equal to the length of the connecting rod, the distance *gh* being equal to *ad*. Striking this arc is, in fact, projecting the point *b* to the centre line with the circular arc instead of a straight line, as has heretofore been done; and in order to find the true relation between the positions of the piston and crank, it is only necessary to project the one to the other by means of such circular arcs with radius equal to the length of the connecting rod. It is often convenient to measure the piston positions for the forward and return strokes from the same starting point as *i*, and in order to do this it is only necessary to strike the arcs for the forward and return strokes from opposite sides of the shaft. Thus, with the crank on the quarter, the piston will have moved

through the distance ih for the forward stroke and ik for the return. Similarly, if the crank move through an angle ibl for the forward stroke, the piston will have moved through the distance io; and if the crank move through the same angle from p on the return stroke the piston will have moved through the distance iq. The directions of these distortions for the two strokes are best distinguished by remembering that the effect of the connecting rod is always to draw the piston too near the crank.

The amount of these distortions will diminish if the length of the connecting rod be increased, and if a connecting rod of infinite length be conceived, the distortions will disappear. Hence a piston motion without distortion, such as is given by the slotted cross-head, is often called the motion due to a connecting rod of infinite length.

Since the speed of the crank's rotation is uniform, and the piston must travel farther for a given angle of crank rotation in the forward than in the return stroke, it follows that the speed of the piston's motion is greater in the forward than in the return stroke.

These principles can be applied to the problems already given, and thereby determine the actual positions at which the various events occur. Fig. 26 is a reproduction of Fig. 22, but with the projections made by circular arcs instead of straight lines. It thus appears that with the valve there designed, the cut-off, instead of taking place as intended in Fig. 22, will really take place after a piston travel Aa', Fig. 26, in the forward stroke and Aa'' in the return. Similarly, the compression will take place after travels Ah' and Ah'', and the release after Ab' and Ab''. If preferred, the construc-

tion may be made by repeating the lap circles below the centre line in position for the return stroke, as is

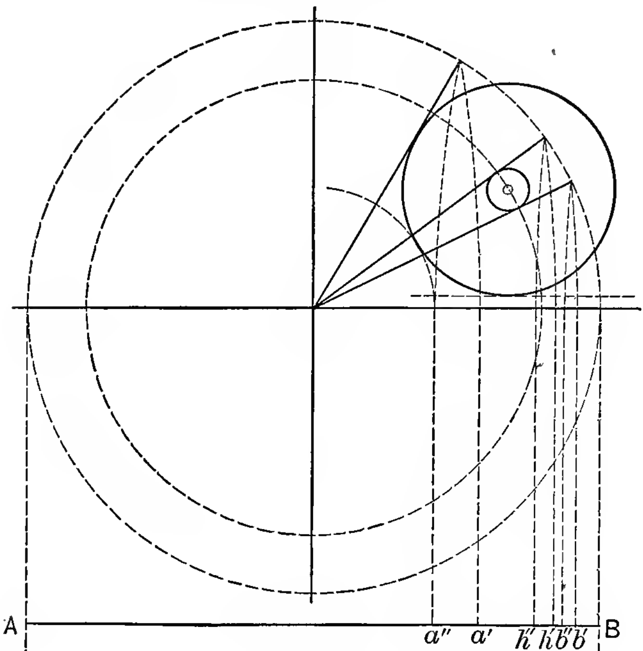

Fig. 26

done in Fig. 27. With this construction the measurements are necessarily made from *A* for the forward stroke and *B* for the return. Of these two plans that of Fig. 26 possesses the advantage that it shows at a glance the difference between the points of cut-off, etc., in the two strokes.

Problem VI. It is required to find the true positions for cut-off, release, and compression of the valve of

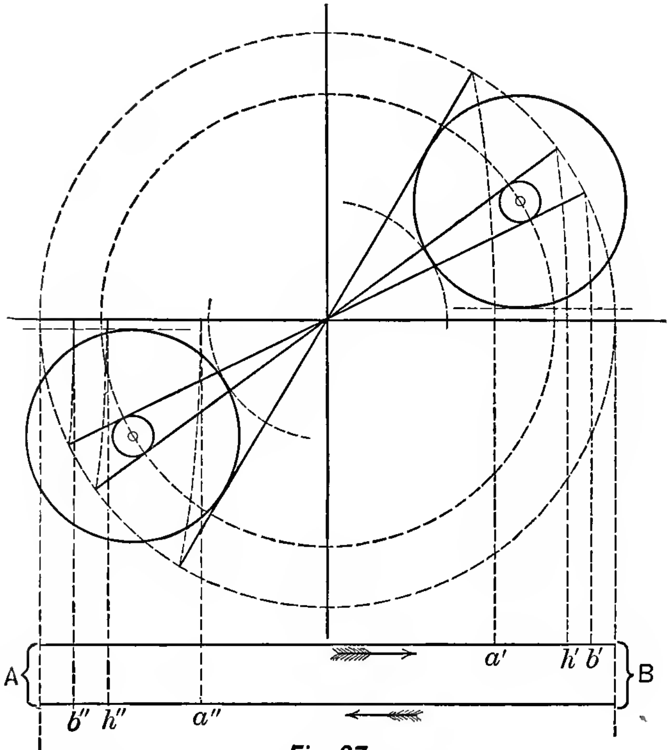

Fig. 27

Problem III. Length of connecting rod five times the crank.

* THE ANGULAR VIBRATION OF THE ECCENTRIC ROD.

As has been explained, the eccentric is in effect a crank and the distortions introduced by the connecting rod into the motion of the piston are likewise introduced by the eccentric rod into the motion of the valve.

In other words, if the distortions are not corrected, the valve, like the piston, will always be too near the crank. The throw of the eccentric is much less than the arm of the crank, and the eccentric rod is proportionately longer than the connecting rod; hence the distortions in the positions of the valve are absolutely and relatively smaller than those in the positions of the piston. Since the effect of these distortions is to draw the valve too near the crank, it follows that if they are not provided for, the lead of the valve at the back end † of the cylinder will be increased and for the front end diminished. The greatest port openings are measured with the eccentric on the centre line of the engine, when these distortions vanish; and hence the two port openings will be equal. It is important that the lead openings be equal, while it is not particularly important that the maximum port openings be equal, provided the smaller one be large enough. Hence in practice the eccentric rod or valve rod is slightly lengthened to give equal lead at the two ends, and the result is that the port opening is slightly diminished for the forward stroke and slightly increased for the return. This lengthening of the eccentric rod is effected in setting the valve for equal lead, and it practically corrects the effects of the angular vibration of the eccentric rod.

* EQUALIZED EXHAUST.

As has been explained, the setting of the valve for equal lead practically neutralizes the effect of the an-

† With apologies to the locomotive fraternity, the end of the cylinder farthest from the crank will be called the back end.

gularity of the eccentric rod. Nothing, however, has yet been done toward correcting the irregularities due to the connecting rod, and that is the next subject to be discussed.

If the valve has no inside lap, compression and re-

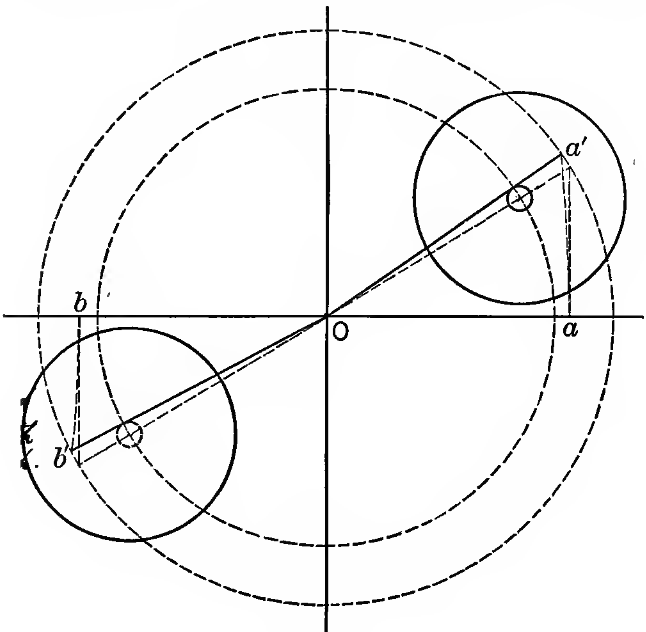

Fig. 28

lease are coincident, and the correction of one will likewise correct the other. It is desired to cause both these events to occur earlier in the forward stroke and later in the return stroke, and to accomplish this, it is only necessary to give an appropriate exhaust lap to the end of the valve nearest the crank shaft, and an equal nega-

tive exhaust lap to the other end. In applying this correction, it should be remembered that the Bilgram diagram gives the true relation between the positions of the crank and eccentric, and that the distortions under discussion are given to the piston through the connecting rod. In Fig. 28 it is proposed to correct the release and compression of the valve shown, which, in the first instance, has no exhaust lap. By the vertical projection lines the points of release a, b are found in the usual way, and by means of the curved projection lines points a', b' are found for the correct crank positions corresponding to piston positions a, b. If the release and compression are to take place at piston positions a, b, they must take place at the crank positions a', b'. Drawing the crank lines $a'O$ and $b'O$, it is easy to add the exhaust lap circles shown, from which measurement shows that for that edge which effects exhaust at a' a ~~positive~~ neg. lap of $\frac{1}{16}''$ is required, and for b' a ~~negative~~ positive lap of the same amount. The resulting valve is shown in Fig. 29. Had

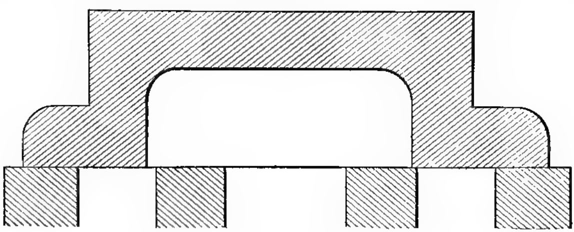

Fig. 29

the valve originally possessed exhaust lap, as in Fig. 30, exact equalization would have been impossible, although a result could have been reached sufficiently nearly correct for all practical purposes. The piston positions

for compression a, b and release c, d are projected to the crank circle by circular arcs, as shown, giving the corresponding crank positions a', b', c', d'. It is apparent at once that the change of lap to give compression at a' is greater than the change to give release at d', and simi-

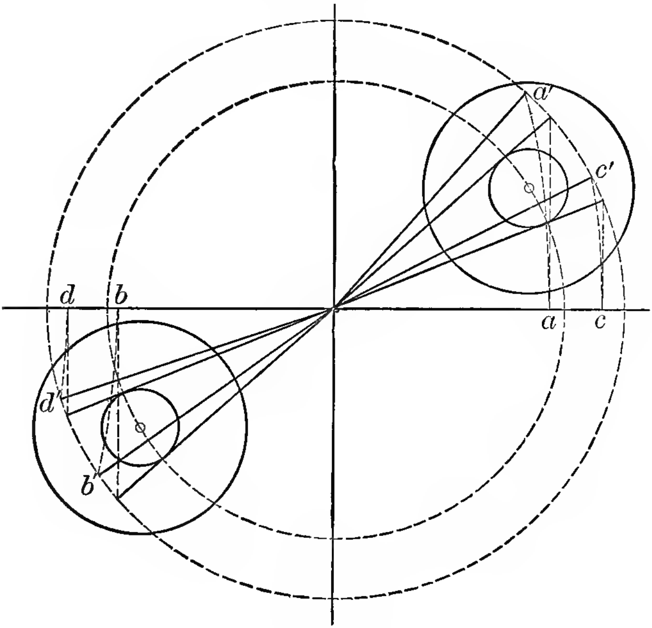

Fig. 30

larly for c' and b'. In such a case the best that can be done is to divide the difference, making the alteration in the lap half way between that called for by a' and d' for their end of the valve, and half way between that called for by c' and b' for their end of the valve. This subject will be returned to at the close of the next section.

omit this article.

* EQUALIZED CUT-OFF.

It was shown in the last section that by introducing inequality in the inside laps, the inequality of release or compression could be equalized—a change in one event being, however, accompanied by a change in the other. It is obviously possible to equalize the cut-off in a similar manner by making the outside laps unequal. As a change in the inside laps involved both release and compression, so will a change in the outside laps involve both admission and cut-off; and since the valve, as thus far described, gives equal lead at the two ends of the cylinder, it follows that increasing one lap and decreasing the other would result in an unequal lead—in other words, cut-off equalized by such a method would involve unequal lead. Such a method is usually explained in detail in books of this character. Equality of lead is, however, of more importance than equality of cut-off,† and hence the method is of no practical importance, and is not introduced here.

The following method secures equality of cut-off without affecting the equality of the lead. It has no objectionable features, and is of general utility. Throughout the discussion, one fundamental fact must be kept in mind, viz.: The acts of opening and closing a port by a slide valve differ only in the direction of motion of the valve. The port is opened or closed, as the case may be, by the edge of the valve passing the

† Except in vertical engines where the lead for the lower centre is usually made larger than for the upper to compensate the action of the weight of the reciprocating parts.

edge of the port, and the position of the valve when cut-off takes place is the same as when admission takes place. Since the valve is mechanically connected to the eccentric rod pin, it follows that the position of that pin must be the same at cut-off as at admission.

Let it be proposed to design a slide valve to cut off steam at half stroke, with equal lead and cut-off.

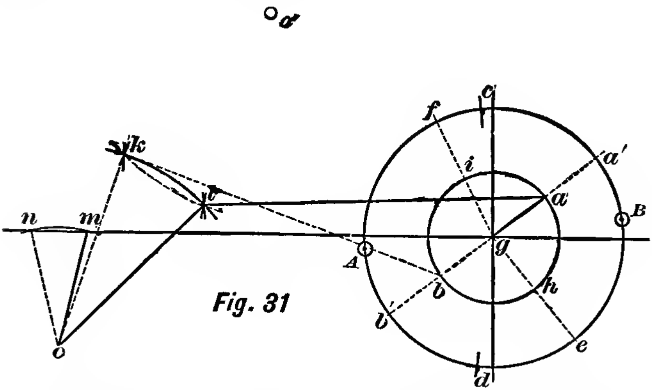

Fig. 31

First design the valve by the methods already explained, then strike the crank and eccentric circles of Fig. 31.* Locate points *A*, *B*, the positions of the crank pin for admission of steam, and the corresponding positions *a*, *b* for the eccentric centre. With radius equal to the length of the connecting rod, and with

* In this and the following diagrams the throw of the eccentric is made disproportionately large, and the eccentric rods disproportionately short, to add to the clearness of the constructions without unnecessarily large diagrams. This gives the appearance of a distorted valve movement; but with working proportions these apparent distortions are no greater than with the usual construction, and are not objectionable.

centre at the middle position of the cross-head pin, strike arcs cutting the crank circle at *c* and *d*. Before cut-off in the forward stroke, the crank shaft, and with it the eccentric, must turn through the angle *Ac*, and in the return stroke *Bd*. Space off *a'e* equal to *Ac*, and *b'f* equal to *Bd*. Draw *eg* and *fg*, and we have point *h*, where the eccentric centre must be for cut-off at *c*, and *i* where it must be for cut-off at *d*. With radius equal to the length of the eccentric rod, and with centres at *b*, *i*, strike arcs meeting at *k*, and with same radius and centres *a*, *h*, strike arcs meeting at *l*. Now for admission at *A* and cut-off at *c*, the eccentric rod pin must be in the same position, and as the eccentric rod is of fixed length, this position must be *l*, that being the only point whose distance from both *a* and *h* equals the length of the eccentric rod. Similarly for admission at *B* and cut-off at *d*, the pin must be at *k*. The pin can be brought to these positions at the proper time by introducing a rock shaft in the valve motion having its centre at any point *o*; such that an arc struck from it shall pass through *k* and *l*, and then connecting the eccentric rod to it as shown. The valve stem should then be connected to the rocker, as shown at *m*, *n*. The eccentric rod positions for crank positions *A*, *B* are shown at *al*, *bk*. The rocker fulcrum might be located above the centre line, if preferred, at *o'*.

In cases where the valve chest is located on the top of the cylinder, a rocker of different type, with the arms on opposite sides of the fulcrum, becomes necessary. The construction for this type of rocker is essentially the same as shown in Fig. 32, which is lettered to correspond with Fig. 31. Of course, with this type of

rocker the eccentric positions *a*, *b* change places as shown. One result of the equalization growing out of the inequality of the rocker arms is to alter the port opening from that determined upon in the original designing of the valve. The motion as thus far determined should therefore be treated as a trial result only, and the dimensions of the valve and eccentric should be altered in the light of the experience gained.

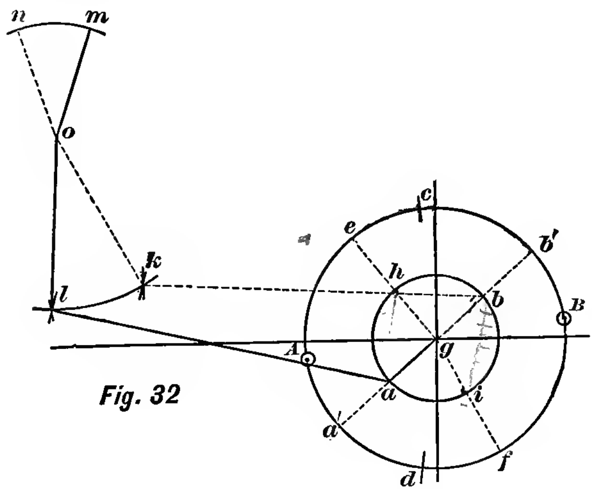

Fig. 32

At the close of the description of Fig. 31 it was stated that the rocker fulcrum might be located indifferently at either *o* or *o'* of that figure. This is strictly true so far as relates to equal lead and cut-off, but there is still a difference in the effect of the two positions. By suitably locating the fulcrum the compression and exhaust can be equalized for the two ends of the cylin-

der—exactly if the valve have no inside lap, and approximately if it have such lap. This has not the unique interest which belongs to the equalization of lead and cut-off, since it can be accomplished by other means; but it forms an interesting study, nevertheless. The method of accomplishing this equalization is shown in Fig. 33, which follows the construction of Fig.

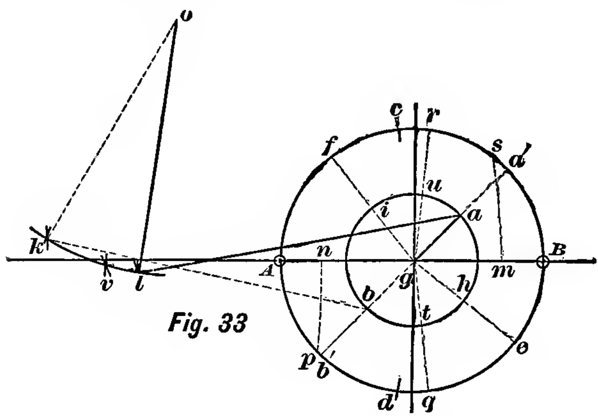

Fig. 33

31 up to and including the finding of *k*, *l*, but with lead zero. Suppose, in the first instance, that the valve has no inside lap, and by the methods already described find the points of the piston stroke *m*, *n*, where release and compression should occur, and by arcs whose common radius is equal to the length of the connecting rod find the corresponding crank positions *s*, *p*. Lay off *Acs* from *a′* giving *q*, and *Bdp* from *b′* giving *r*. Draw *qg* and *rg* giving *t* and *u*, where the eccentric must be for the two equalized compressions. With radius equal to the eccentric rod, and centres *t*, *u*, strike arcs meeting in

v. Now locate the rock shaft fulcrum at *o*, such that the eccentric rod pin shall pass through *k*, *l*, and *v*, and the result will be a valve motion giving equal lead, cut-off, release, and compression. If the valve have inside lap, then, instead of one point, *v*, there will be two, just as with outside lap there are two points, *k*, *l*. In that case it will be found impossible to so locate *o* that the eccentric rod pin shall pass exactly through all four points. It should be then made to pass through *k* and *l*, and the difference be divided between the two points *v*. The release and compression will then be as nearly equalized as is possible.

SETTING THE SLIDE VALVE.

As the parts of an engine valve motion are assembled two dimensions are lacking: 1st, the angular location of the eccentric relative to the crank; and, 2d, the length of the valve rod. The eccentric is capable of being located in any angular position, and the length of the valve rod is usually capable of adjustment by means of jamb nuts each side of the valve, or some equivalent means. The setting of the valve involves locating the eccentric and fixing the valve at the proper point on the rod. There are two distinct steps to the process:

I. Locating the engine exactly on the centre;

II. Locating the eccentric and valve.

To locate the engine on the centre, proceed as follows: Turn the crank to any convenient distance above the centre, Fig. 34. Upon the side or face of the crank disc or fly-wheel, as most convenient, scribe an arc *a* by means of a tram *b* swinging from any conven-

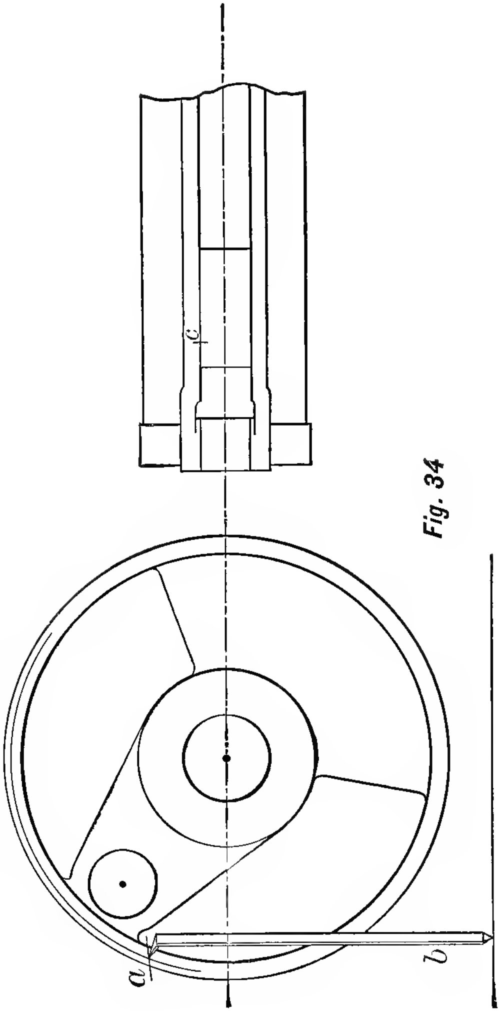

Fig. 34

ient fixed point on the engine frame or floor. Also scribe a line *c* on cross-head and guides. Turn the crank below the centre as shown by Fig. 35, the cross-head line receding from its mate on the guide and approaching it again. When these lines are exactly fair, stop the motion and scribe a second line *d* on the wheel, line *a*, now occupying the position shown in Fig. 35. With the dividers find point *e*, dividing the arc *ad* in halves. When point *e* is brought fair with the point of the tram, Fig. 36, it is clear that the engine will be on the centre. Repeat this construction for the other centre. One precaution is necessary in relation to the above, in order to obviate any error that might arise from looseness in the crank pin and cross-head pin bearings: In scribing the lines *a* and *d* have the crank pin pressing against the same brass for both lines. It matters not which brass be used, but the same one must be used for both lines. To locate the eccentric and valve proceed as follows: Locate the eccentric by the eye as near as may be, and ahead of its correct position rather than behind it. Bring the engine to either centre, as found above, *turning it in doing so in the direction of the proposed rotation* in order to neutralize any looseness in the connections. With the engine on the centre, locate the valve to give the required lead, after which turn the engine *in the direction of its future rotation* to the opposite centre. If the eccentric is ahead of its correct position, the lead for this position will be greater than the first; if the eccentric is behind, the second lead will be less than the first, and probably negative. In either case the valve is to be adjusted on the rod to divide the differences in the lead. This being done, the valve is cor-

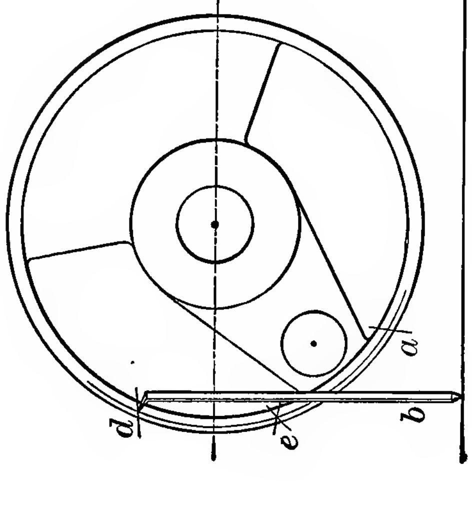

Fig. 35

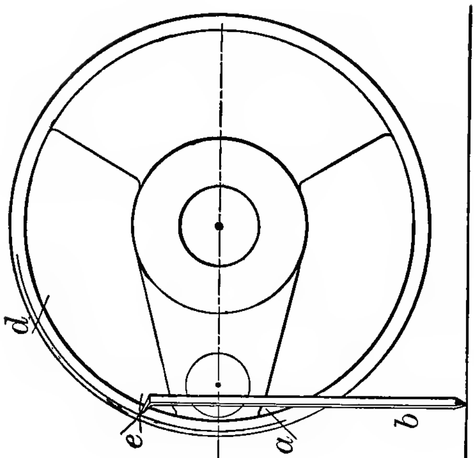

Fig. 36

rectly located on the rod, the lead is equal at the two ends of the cylinder, but is too large or too small at both. To correct this it only remains to adjust the eccentric, *moving it in the direction of the rotation* until the valve have the proper lead. Verify the results, and the work is done.

April 15 '10 – –

Part II.

THE SLIDE VALVE WITH SHIFTING AND SWINGING ECCENTRIC.

The Slide Valve with Shifting and Swinging Eccentric.

THE SLIDE VALVE AT SHORT CUT-OFF.

The difficulties which impede the use of the plain slide valve at short cut-off have been explained at length in Part I. Before explaining the shifting eccentric automatic valve gear, it is necessary to show how these difficulties have been surmounted. By referring to the section on the Limitations of the Plain Slide Valve those difficulties will be seen to be—

I. Premature release and compression—either of which, however, can be made later at the expense of making the other earlier still.

II. Inadequate port opening to steam or, in lieu of that, excessive size and travel of valve.

The first difficulty has been met by increasing the speed of the engine. All of the engines employing this description of valve gear are of the "high speed" type. In such engines a heavy cushion is appropriate and necessary to bring the reciprocating parts quietly to rest at the centres, and hence the early compression ceases to be a radical objection. Indeed, inside lap is

given to the valve in order to delay the release—thereby, as has been explained, still further increasing the compression.

The second difficulty is met by two expedients, the first being sometimes employed alone, but more often in connection with the second. These expedients are, 1st, the use of balanced valves, usually of the true piston type or of the "pressure plate" type, both being perfectly balanced against the steam pressure; 2d, the use of valves having multiple ports, by which the necessary throw of eccentric is halved or even quartered. The use of balanced valves permits the use of valves of large size and great throw; and the use of multiple ports gives sufficiently large openings with such throws as it is practicable to use.

It is believed that the first engine to embody the above features in connection with a shifting eccentric and a shaft governor was the Straight Line; and hence that engine is entitled to be recognized as the progenitor of a large and vigorous family. So far as known, these features were first combined in an engine designed by Professor John E. Sweet, built at the Cornell University shops, and exhibited at the Centennial Exhibition.

That the difficulty of restricted port opening is a real one, may be gathered from any indicator diagram from a locomotive with plain valve at good speed and well "notched up." Such diagrams invariably show a marked fall in the steam pressure on entering the cylinder; and it is largely on this account that such persistent attempts have been made to improve the locomotive valve motion. An appropriate introduction to

the study of multiple ported valves is found in one not necessarily balanced, which was designed more especially for use on locomotives,—to which it has been largely applied,—namely, the Allen valve, shown in Fig. 37. In this valve the seat is shortened and a supplementary port *aa* is cast through the valve. This port registers with the end of the seat as shown in the figure, which represents the valve open by the amount of its lead. The course of the steam is shown by the arrows, from which it will be seen that the opening at *b* is added

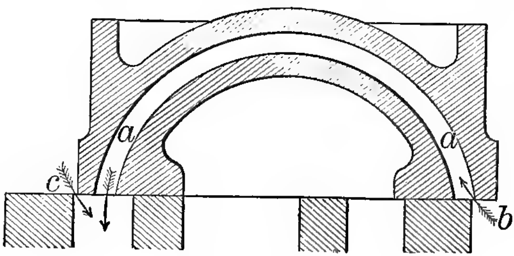

Fig. 37

The Allen Valve.

to the usual one at *c*; and that up to the point where the opening at *c* is equal to the width of passage *a*, the total opening is just twice what it would be with the usual form of valve.

The "pressure plate" type of valve is well shown in Fig. 38, which represents the valve of the Straight Line engine. The pressure plate *AA* receives the pressure of steam upon its back. It is prevented from pressing the valve proper to its seat by means of distance pieces above and below the valve, and slightly thicker than the valve. Recesses in the plate form in it an exact coun-

terpart to the valve seat. The valve slides between the seat and plate like a square piston relieved of all pressure. This valve, like those that follow, is shown

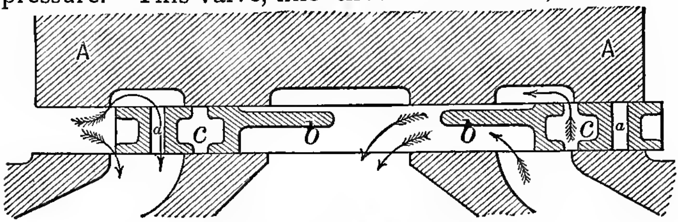

Fig. 38
The Straight Line Valve.

open to its lead ; and the manner in which the recesses in the plate and the passages *aa* through the valve combine to give a double port opening will be seen

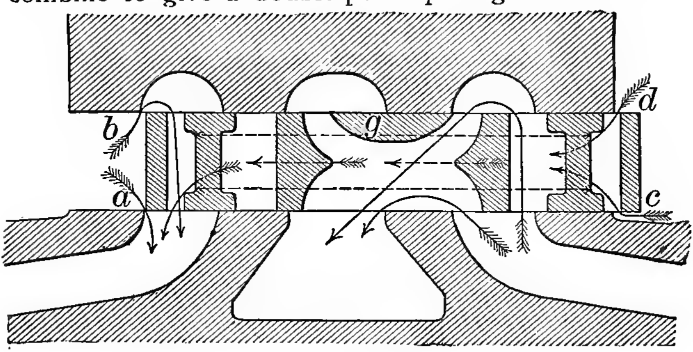

Fig. 39
The Woodbury Valve.

from the arrows. The ledges *bb* are for the purpose of protecting the finished surfaces of the pressure plate from the cutting action of the exhaust steam. Some designers of double ported valves have thought it best to provide double ports for the exhaust, while others

have not. The illustrations of both the Allen and Straight Line valves will show that the opening to exhaust is amply large without double ports; but such ports increase the *quickness* of opening to exhaust, and so secure a desirable advantage. The valve under discussion is provided with them at *c*, *c*. Their action is

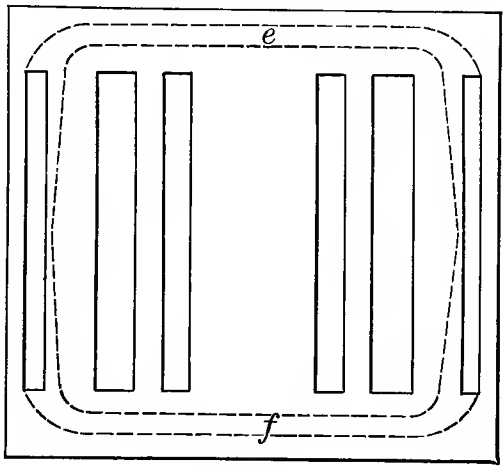

Fig. 40

precisely the same as that of the steam passages *a*, *a*, and need not be explained further.

Fig. 39 shows the valve of the Woodbury Engine Company. It combines the method of action of the Allen and Straight Line valves, and so secures four port openings to steam and two to exhaust. Openings *a*, *b* act precisely like those of the Straight Line valve, and openings *c*, *d* act substantially like the supplementary port of the Allen valve. This will be seen more clearly by re.

ferring to Fig. 40, which represents a plan of the Woodbury valve. Passages *e, f* are cast through the valve to act in conjunction with the openings *c, d* of Fig. 39, in the same manner that the passage *aa* of Fig. 37 operates in conjunction with opening *b* of the same figure. Ledge *g* acts to protect the finished surfaces of the cover plate from the action of the exhaust in the same manner as ledges *b, b* of the Straight Line valve.

Fig. 41 illustrates the valve of the Armstrong engine,

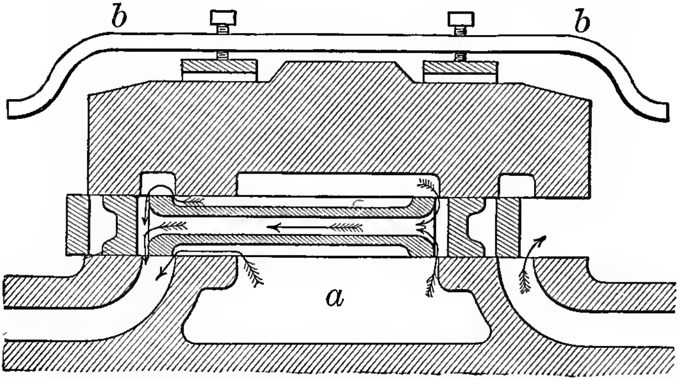

Fig. 41
The Armstrong Valve.

in which the action of the steam and exhaust edges of the valve is reversed from the usual practice. The steam enters at *a*, and the outer edges of the valve control the exhaust. The steam pressure tends to lift the cover plate, and it is therefore held down to its seat by means of the bridle *b*. The action of the steam ports will be seen from the arrows. This valve gives four openings to steam and two to exhaust.

The Rice valve (Fig. 42), like the last example, takes

steam from the inside. It gives two openings to steam and two to exhaust. The relief plate *aa* is in this instance a piston fitting the cylinder *bb*, this cylinder be-

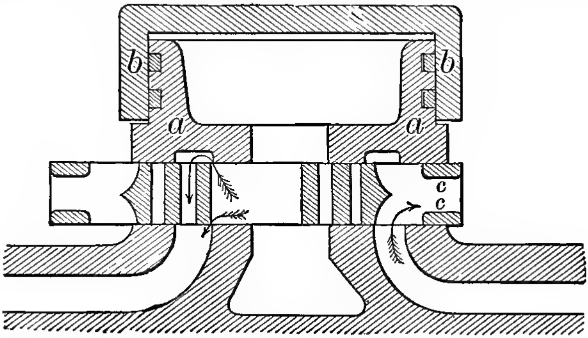

Fig. 42
The Rice Valve.

ing bolted to the floor of the steam chest. The pressure of steam within this cylinder forces the piston toward the valve, which, however, it is prevented from

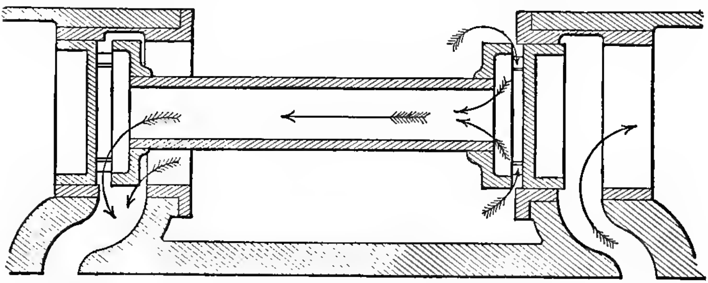

Fig. 43
The Armington and Sims Valve.

touching by means of distance pieces slightly thicker than the valve, and similar to those already described in connection with the Straight Line valve.

Fig. 43 illustrates the Armington and Sims valve, which is a true piston valve with double ports. The course of the steam is shown as heretofore by the arrows.

Fig. 44 shows the Ide double ported valve, which, like the last, is a piston valve.

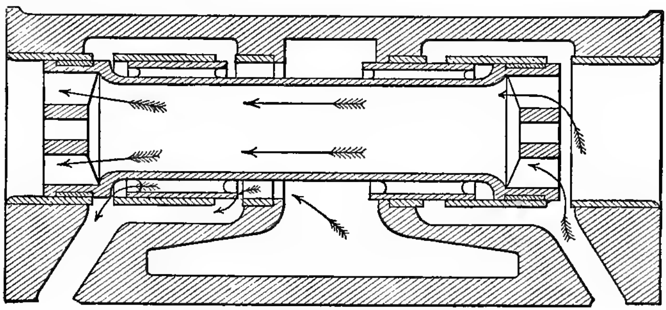

Fig. 44
The Ide Valve.

Of an entirely different type is the Giddings valve as used in the Russell & Company engine. This valve is shown in Fig. 45. Steam enters at *a* and exhausts at *bb*. Each end of the valve acts in much the same manner as the Allen valve, as will be understood from the arrows, while over all is cast a case *c*. The steam entering from the inside of the valve, its pressure would, if not counteracted, lift the valve from its seat. This is prevented by the use of "needle ports" (not shown), one connecting the live steam space within the valve to the body of the valve chest, and the second connecting the chest to the exhaust. The action of these ports is explained by Mr. Giddings as follows: "The steam is taken under the valve, which would result in throwing the valve off

the seat. This must be counteracted by pressure on the back of the valve sufficient to overcome the tendency to leave the seat. This I obtain by the small needle port communicating with the live steam passage on the inside of the valve. If there were no outlet this would soon result in an excess of pressure nearly equivalent to steam pipe pressure on the back of the valve, which would produce a hard working valve. To avoid this, I put another needle port opening, communicating with one of the exhaust 'D's' of the valve.

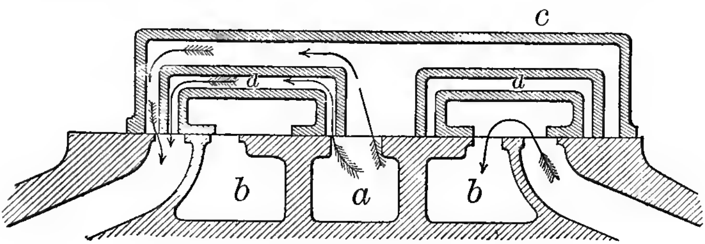

Fig. 45
The Giddings Valve.

The resulting pressure due to these two openings is just sufficient to overcome the tendency to leave the seat."

A remarkable feature of this valve, not attained by any other so far as known to the author, is the use made of the supplementary passage *d*. After the compression has commenced, and before opening to admit live steam from the steam supply, this passage opens into communication with the regular steam port. This increases the volume into which the steam is compressed, without, however, increasing the clearance space from which the steam is exhausted, since this supplementary

port is never in communication with the exhaust. This is described by Mr. Giddings as follows :

" We find by increasing the capacity of the carry over port or portchamber, that we can use it as a reservoir or port into which to pack the surplus compression of a single valve automatic movement, thereby giving us a peculiar offset in the compression curve, and giving us from 10 to 12 per cent increase of area, and a consequent increase of power from a given sized engine. The amount of this is entirely within our control by a

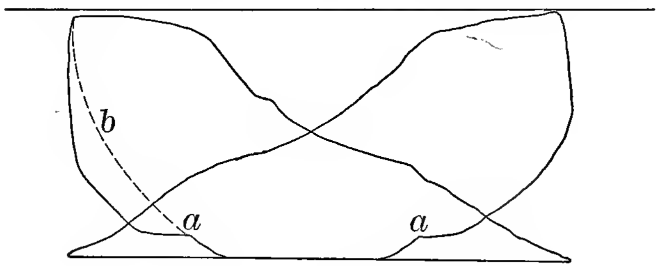

Fig. 46

variation of the cubical capacity of this passage way, thereby enabling us to get the compression curve down in the corner, something after the manner of the four valve engine cards."

This feature is of such unique interest that indicator-cards from one of these engines (Fig. 46) are introduced to illustrate it. The offset mentioned will be seen at *aa*, the dotted line *b* representing the compression curve that would have resulted but for this provision.

The shifting eccentric valve gear, in connection with which all these valves are used, requires that the valve

shall have an excess of port opening and travel at the late cut-offs in order that they may be sufficient at the early cut-offs. Inspection of any of the valves illustrated will show that with large travels the supplementary port becomes closed after the main port is well opened; and with such travels the effect of the supplementary port is merely to increase the quickness of port opening and closing.

* EQUAL LEAD AND CONSTANT LEAD.

The lead of a valve may vary in two ways, and to prevent ambiguity it is necessary to define and follow an exact use of terms. The reader is asked to note carefully the following explanations of the terms equal lead and constant lead. They will be used hereafter strictly as defined, and it is necessary that they be clearly understood.

Equal lead implies that the lead is alike at the two ends of the cylinder. Constant lead implies that the lead does not change for different grades of expansion. An engine might have equal lead for one grade of expansion and unequal lead for another grade, or the lead might be equal at all grades without being constant; that is, the lead at the two ends of the cylinder might be always alike, but larger at both ends for three quarters cut-off than for one quarter. So, also, an engine might have constant lead without equal lead; that is, the lead might not change for different grades of expansion, but at the same time be always larger for one end of the cylinder than for the other.

The above distinctions are radical and important,

and it is necessary that they be clearly seen in order to understand what follows.

THE SHIFTING ECCENTRIC.†

The expedients which are employed for making what is geometrically the plain slide valve available for early cut-offs having been explained, it remains to describe how the same valve can be made available for different cut-offs.

In Fig. 47 let the circle *abc* represent the eccentric path of a given eccentric in the Bilgram diagram, the eccentric centre being at *d*, and *L* and *l* being, as usual, the steam and exhaust lap circles. Let the centre of the eccentric be shifted from *d* to *d'*, *dd'* being a straight line perpendicular to *ac*. There will thus be formed a new eccentric path *a'b'c'*. Laying off *b'd'* upward from *c'*, will locate *Q'*, the new centre of the lap circles. From this centre the lap circles may be drawn in the usual way, and from them the crank positions *A* for the new point of cut-off, *B* for exhaust closure, and *C* for release may be found. Since *bd* equals *cQ*, and *b'd'* equals *c'Q'*, and since *dd'* is perpendicular to *ac*, it follows that *QQ'* is parallel to *ac*: hence the lead opening has not been changed. The lead angle, however, has clearly been increased, and the port opening has been reduced. In the same way, the eccentric may be shifted still further on the line *dd'*, and new points of cut-off, release, and compression, and new values of the

† Throughout this and the following section the angularity of the connecting and eccentric rods is neglected.

port opening, found. Finally, if the eccentric be shifted to the position $d°$ on the line ac, the point Q will be located at $Q°$, when the port opening will be reduced to the lead opening and the cut-off will take place as much after the centre as the lead did before it. In all posi-

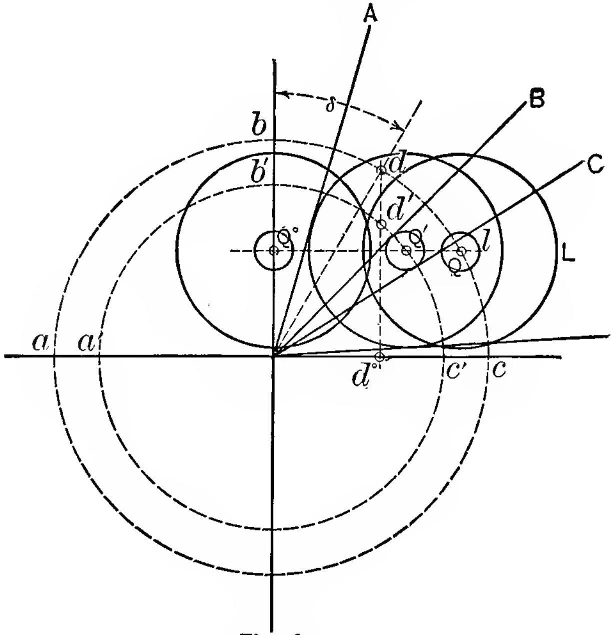

Fig. 47

tions the lead opening will be constant. If the valve have multiple ports, when the travel becomes so reduced as to bring them into action, the result will be to give double or quadruple the opening shown by the diagram, as the case may be. Should the movement of the eccentric be continued below the line ac, the result would

be to reverse the direction of the rotation. The study of reversing engines is, however, beyond the scope cf the present work.

An eccentric arranged for adjustment on straight line dd°, as in this illustration, is called by the author a *shifting* eccentric.*

So far as known to the writer, there are but two engines in the American market which employ a shifting-eccentric. These are the Armington and Sims and the Russell (Giddings) engines. The former obtains practically a straight line motion of the eccentric centre by means of a combination of two eccentrics, while in the Russell engine the required motion is obtained by means of a straight guide keyed to the shaft and appropriate wings attached to the eccentric.

THE SWINGING ECCENTRIC.

Instead of shifting the eccentric across the shaft in a straight line as in the last examples, most designers have preferred to swing it by an arm cast in one with itself, and pivoted to an arm in the fly-wheel or other convenient piece. Such an eccentric is called by the author a *swinging* eccentric, and its effect upon the steam distribution, as distinguished from the shifting eccentric, is to vary the lead opening at different points of cut-off. The nature and degree of this variability depends in large degree upon the location of the pin from which

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* For want of another word, the term *shifting eccentric* is also used as a general expression which includes both shifting and swinging eccentrics. This double use of the word will not, however, cause confusion.

the eccentric is swung. If the pin is on the same side of the shaft as the crank, and in the centre line of the crank, as in Fig. 48, the path of the eccentric across the shaft is the arc *dd*°. The path of the point *Q* will then be a similar arc *QQ*°, with the same radius and with its centre located in the line *ef*. It is clear from the positions of the various lap circles that the lead opening

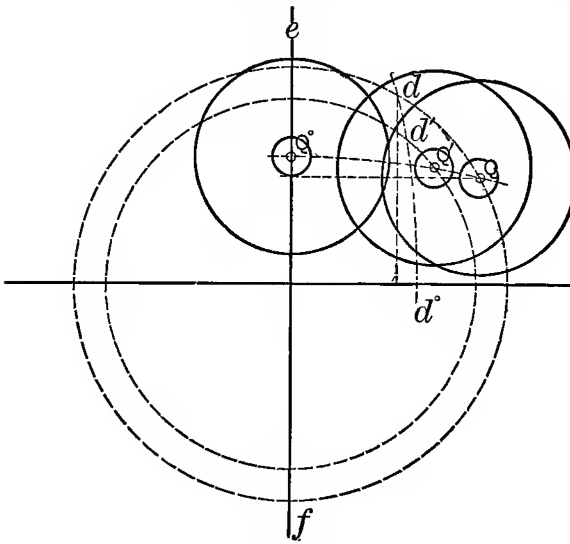

Fig. 48

will not be constant with this arrangement, but will be greater in the early cut-offs than in the late ones. The action upon the other events of the stroke is substantially the same as that of the shifting eccentric. With the pin located in the centre line on the opposite side of the shaft from the crank, as in Fig. 49, the arc in which the eccentric swings has its convexity reversed

from the last figure. The path of the point Q will als(be reversed, and instead of receding from the centre line in the early cut-offs, it will approach it, thus diminishing the lead opening in those cut-offs. This diminution of lead may be carried so far as to make the lea(zero or even negative in the early cut-offs—a fact whicl

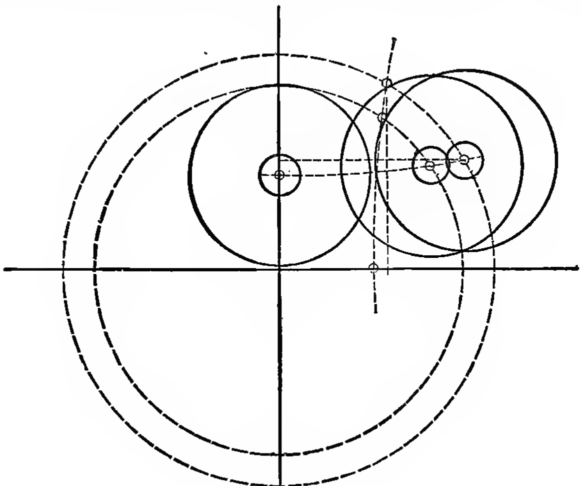

Fig. 49

will be shown on the diagram by the earlier steam la] circles crossing the horizontal centre line. The sam(result of a decreasing lead in the early cut-offs can b(obtained with the eccentric swing pin on the same sid(of the shaft as the crank, but raised above the centr(line as in Fig. 50, in which the swing pin is located o] the line dg extended. Similarly, by raising this pi] when located opposite to the crank, an increasing lea(can be obtained. By locating the pin half way betwee]

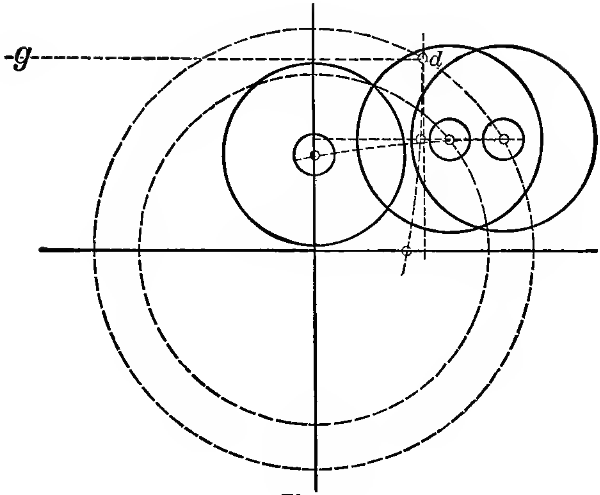

Fig. 50

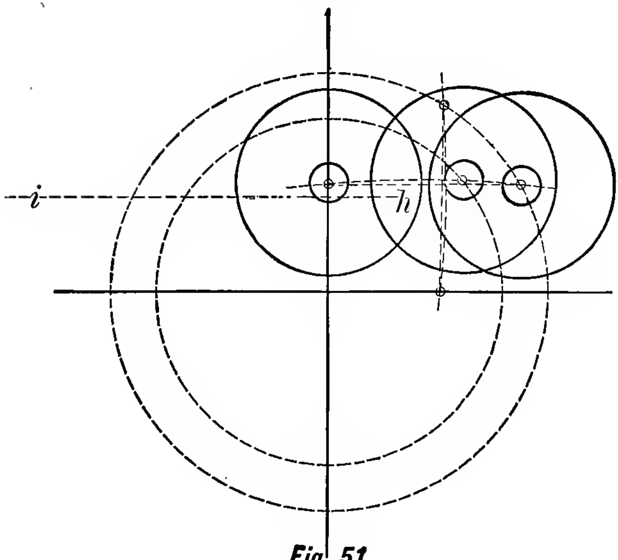

Fig. 51

the positions of Figs. 48 and 50, on the line *hi* of Fig. 51 the lead will be the same at the smallest as at the greatest throw; and by suitably placing it as in Fig. 52, the lead can be made alike for any two expansions desired. In this construction the two points of cut-off

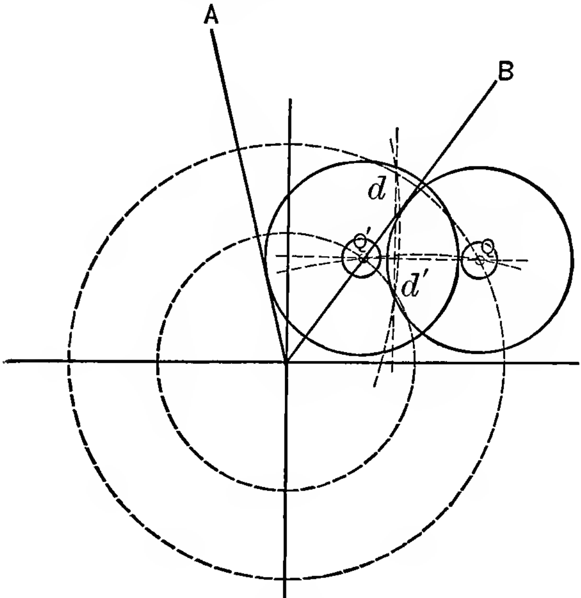

Fig. 52

for which equal lead is desired are decided upon, and the corresponding crank positions *A*, *B* are drawn. The lap circles are drawn in the usual way for *A* and *B*, the lead being made the same for both. Points *Q* and *Q'* are then transferred to *d* and *d'*, and the eccentric is so hung that its centre shall pass through the points *d* and *d'*.

As has been pointed out, if the valve have a negative lead in the early cut-offs, it will be shown in the diagram by the lap circle going below the horizontal centre-line. In this case if the cut-off be made early enough the lap circle will pass through the centre of the shaft. This marks the point where the port opening becomes zero by reason of the eccentric throw becoming reduced to equality with the lap. For any smaller throw there is no admission whatever.

Designers have usually endeavored to obtain as nearly a constant lead as possible. The author considers, however, that for stationary engines, where the speed is fixed, a lead decreasing in the early cut-offs is more suitable. That the lead should be equal at the two ends of the cylinder, there is, however, no question. This entire subject will be discussed at greater length in the section on Equalized Lead.

* THE ANGULARITY OF THE ECCENTRIC ROD.

Thus far in Part II the influence of the angular vibration of both connecting and eccentric rods has been ignored. It will be understood that primarily both connecting and eccentric rods produce the same distortions with the shifting as with the fixed eccentric. There is, however, with the shifting or swinging eccentric an additional distortion produced by the angularity of the eccentric rod growing out of the fact that the vibration of that rod varies in amount with the varying throw of the eccentric—the small throw due to an early cut-off giving a small angular vibration, and the large throw due to a late cut-off giving a larger vibration.

Taking up first the case of the shifting eccentric in Fig. 53,* let *A*, *B* represent the crank pin of a shifting eccentric engine when on the dead centres, *a*, *b* being the corresponding positions of the eccentric centre at its greatest throw, *h*, *i* at its mean throw, and *e*, *f* at its smallest throw. The positions of the eccentric rod for mean throw of the eccentric are shown at *ch*, *di*, and the valve in both positions is open by the amount of its lead, as shown by the upper valve sketch for *c*, and the lower one for *d*. Let the path of the eccentric

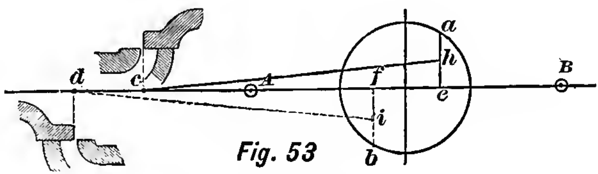

Fig. 53

centre, when shifted across the shaft to change the expansion, be a straight line occupying the position *ae* for crank position *A*, and *bf* for crank position *B*. Imagine the crank on the centre *A*, and shift the eccentric toward *e*. Obviously point *c* will be moved to the left, and the lead will be disturbed. For crank position *B* the same is true; and what is still worse, while the movement for crank position *A* decreases the lead, that for *B* increases it. If, with crank at *A*, the eccentric be shifted towards *a*, and with crank at *B* towards

* See foot-note, page 55.

This feature of the present diagrams shows a greater irregularity in the lead in the usual form of construction, as well as in the form to be described, than actually obtains with working proportions. This, however, for the present purpose, is rather an advantage than otherwise.

b, the lead at the two points will be disturbed in the opposite directions; i.e., for position *A* the lead will be increased, and for *B* decreased. The broad fact is evident, that, owing to the varying angularity of the eccentric rod, an engine laid out as shown in Fig. 53 could not have a constant lead, and it could only have an equal lead for some one (selected) grade of expansion.

With a swinging eccentric, the simplest case is where the eccentric swings about a point which, with crank at *A*, Fig. 54, coincides with *c*. Thus, suppose a

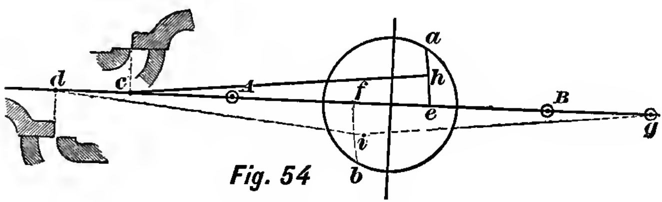

Fig. 54

large disc keyed to the shaft, and arrange the eccentric to swing about a pin fixed to the disc, the centre of the pin for crank position *A* coinciding with *c*, Fig. 54. Now shift the eccentric from *h* towards *e* or *a*, and *c* will not be disturbed. When, however, the crank is at *B*, the pin *c* will be at *g*; and if the eccentric be shifted from *i* towards *f* or *b*, point *d* will obviously be disturbed more than in the corresponding movement of Fig. 53; and it may be said in general, that with the eccentric rod arranged in the common way, as shown in Fig. 53, any change in the path of the eccentric across the shaft, to correct the inconstant lead at one dead centre, will only make matters worse at the other, and by no possible modifica-

tion can the lead be made equal for more than one grade of expansion.

The arrangement of Fig. 54, however, while of interest and value in a theoretical study of the subject, is of no practical importance, because its use would require so large a disc for the attachment of the eccentric swing pin as to be impracticable. It therefore becomes necessary to examine the situation with the eccentric swung from a position nearer the shaft. Let the pin be located at the centre of the crank pin—a common position—as in Fig. 55, the eccentric rod being much longer than the crank, as it always is in practice. In that case the actual path of the eccentric

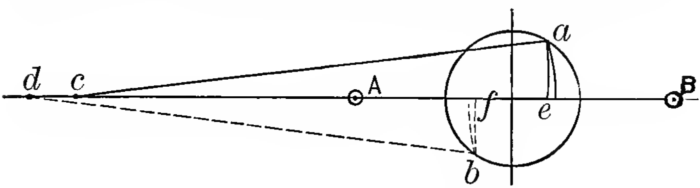

Fig. 55

for crank at *A* will be to the right of *ae* of Fig. 54, and to the left of *bf*; consequently the lead will increase at both ends of the cylinder for short cut-offs, but the increase will plainly be greater for crank position *B* than for *A*. The result is a lead increasing in the early cut-offs, but increasing much faster for one end of the cylinder than for the other, and hence equal at the two ends of the cylinder for one grade of expansion only. Similarly it might be shown that by swinging the eccentric from a point diametrically opposite the crank pin the lead would decrease in the early cut-offs (see Fig. 49), but decrease much more rapidly for one end

of the cylinder than for the other; and it is clear that whatever quality is sought for in the lead, and determined so far as the Bilgram diagram can do it, it will in fact be modified by the angular vibration of the eccentric rod. It seems unnecessary, however, to examine the subject in detail further.

* EQUALIZED LEAD.

So far as known to the author, the only engine in which any attempt is made to correct the distortions which have just been explained is the Straight Line. The valve motion of this engine, like its mechanical details, is an exhibition of refined ingenuity which it would be difficult to surpass. In the following it will first be explained how the engine was originally built to secure a substantially constant lead, and after that the present construction will be shown. The construction to be described is essentially the same as that already used in equalizing the cut-off of fixed-eccentric engines (page 54), and it should be understood that the present use of that construction was the original one—its use for equalizing the cut-off being in fact an offshoot of its original use for equalizing the lead. In equalizing the lead as well as the cut-off, two types of rocker are possible. The first type of rocker is shown in Fig. 56, the parts being lettered as in the three preceding figures. From *h* and *i*, *hc* and *id* are drawn parallel to the centre line, and each is made equal to the length of the eccentric rod. The rocker fulcrum is then so located at *o* that the pin for the eccentric-rod shall describe an arc passing

through *c* and *d*. The pin for the valve rod is located as usual, *m* belonging with *c*, and *n* with *d*. The eccentric is shifted by being swung from a pin, whose location for crank position *A* coincides with *c*. As explained in connection with Fig. 54, the disturbance of the lead for crank position *A* is thus eliminated. When the crank is at *B*, *c* is at *g* in line with *id*. In this position of the parts, shifting the eccentric on the line

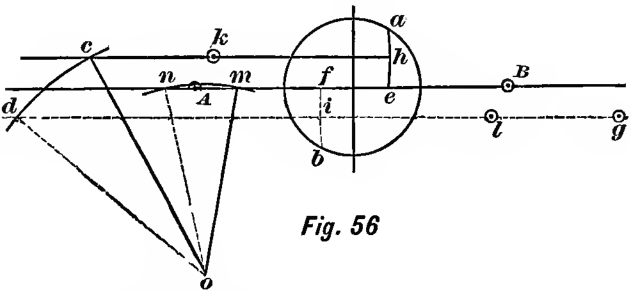

Fig. 56

bf will disturb the lead a trifle, though much less than in the constructions of Fig. 53 or 54. We have here, then, a construction which eliminates the disturbance for crank position *A*, and practically eliminates it for *B*, and thus secures substantially a constant lead. Comparing Figs. 54 and 56, the essential difference in the two plans is apparent. In Fig. 56, *ch* and *di* are parallel, and hence both *c* and *g* are in line with the eccentric rod position to which each belongs; whereas in Fig. 54 *ch* and *di* are not parallel, and hence, while *c* is in line, *g* is not, and cannot be made so. The construction so far explained, would, however, lead to inconvenient dimensions of some of the parts. The centre of motion for shifting the eccentric is therefore in practice moved

inward from c to some convenient point k on the line ch, the eccentric rod pin still remaining at c. The position of k for crank at B is of course l. This change introduces a trifling error for crank position A, and by an equal amount increases the existing error for B; but the final irregularity is infinitesimal as compared with Fig. 53 or 54, and the mechanism accomplishes its ob-

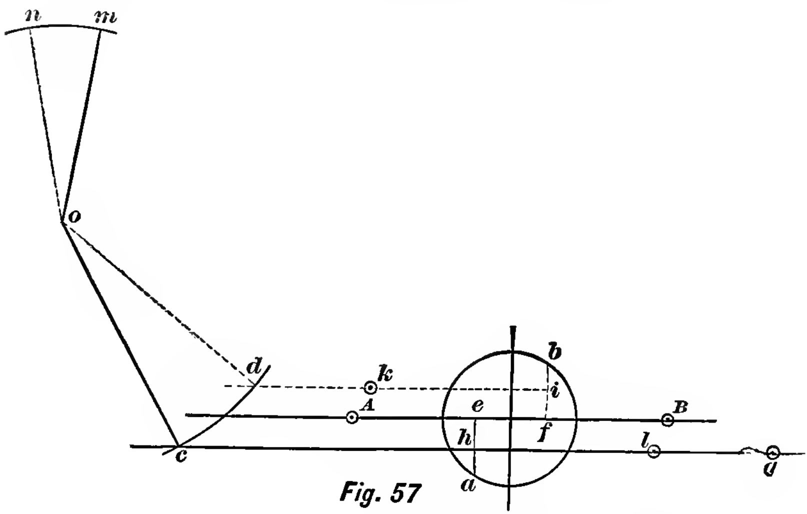

Fig. 57

ject—obtaining a practically constant and equal lead at all grades of expansion.

The second type of rocker introduces no material change in the lay-out. For an engine with the steam chest on top of the cylinder it is illustrated in Fig. 57, the parts being lettered to correspond with Fig. 56. This type of rocker reverses the motion of the eccentric, and hence positions a, b, change places as shown.

As the Straight Line engine is actually built, however, the steam chest is on the side of the cylinder, and hence the second type of rocker takes the unusual form of Fig. 58. There is, however, no change in the essential principle of the construction, which is, that the two positions of the eccentric rods which belong with crank positions *A*, *B* shall be parallel to one another. The object of using this form of rocker is as follows:

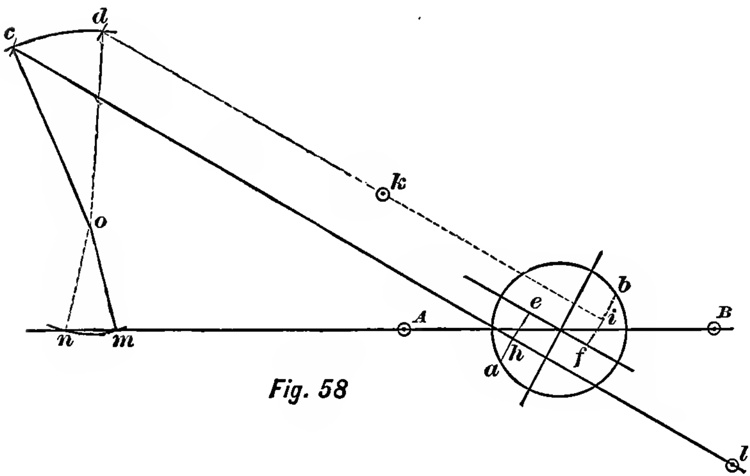

Fig. 58

This valve motion, like the usual form, when set for equal lead, gives a larger port opening for one end of the cylinder than for the other. It is well known that the speed of the piston is faster in the end of the cylinder farthest from the crank shaft. Now the second type of rocker gives the large port opening to that end of the cylinder in which the piston travels the faster,

while the first form gives the reverse relation. Hence the choice in the construction.

As now built, however, the Straight Line engine has a decreasing lead in the early cut-offs, becoming in fact negative in the earliest grades. The reason for this change in practice is as follows: In shifting eccentric engines, as is well known, the compression increases as the cut-off grows shorter. The total cushion by which the momentum of the reciprocating parts is arrested is the sum of the exhaust cushion and the lead cushion. Now if the total cushion is to be constant at all points of cut-off, as it should be, the lead must decrease as the compression increases, and at the early cut-offs the lead should be negative. Furthermore, in such engines, as has been explained, both the release and compression for early cut-offs occur earlier than is desirable. Now by laying out two valves for the same cut-off, one with positive and the other with negative lead, it will be found that the valve with negative lead gives considerably later release and compression than does the one with positive lead. In other words, the introduction of negative lead at the early cut-offs, in addition to offsetting in a measure the increasing compression, prolongs the expansion, thereby getting more work out of the steam, and also delays the compression, thereby still further reducing the cushion. The result is accomplished as shown in Fig. 59, which is a modification of Fig. 58. The point *k*, instead of being on the line *di*, is placed above it. This brings *l* equally below *ch*, extended; and, as will be seen by referring to the upper valve sketch for *m* and the lower one for *n*, reduces the lead for both crank positions *A* and *B*, as the eccen-

tric is shifted inward towards *e* and *f*. If the elevation of *k* above *di* is sufficient, the lead at the early cut-offs will obviously be negative. To fully appreciate the merit of this construction, Fig. 59 should be compared with Fig. 53, when it will be seen that while in Fig. 53 shifting the eccentric inward from *h*, *i*, decreases the lead in the upper valve sketch, it increases it in the lower one; in Fig. 59, on the other hand, shifting the

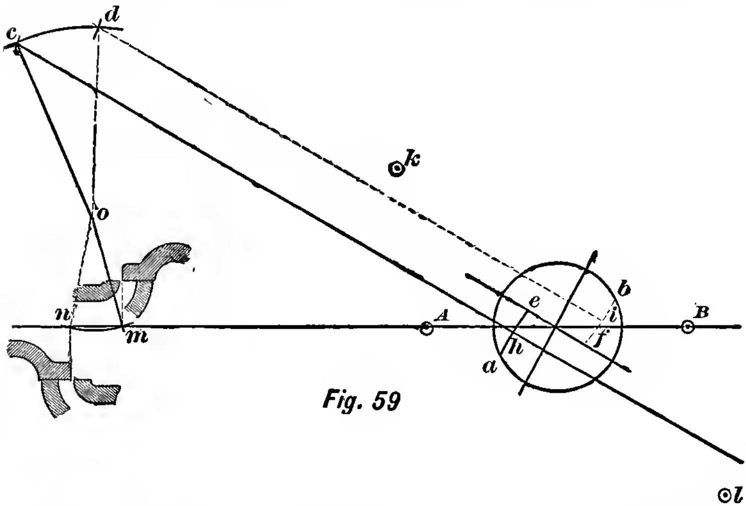

Fig. 59

eccentric inward from *h*, *i*, decreases the lead in both valve sketches.

Again, Fig. 59 should be compared with Fig. 55, and with the suggested companion to it having a decreasing lead in the early cut-offs, when it will be seen that while the plans are essentially alike in having an inconstant lead, they are unlike in that in Fig. 59 the lead is substantially equal at both ends of the cylinder

throughout the range, but in Fig. 55 it is equal at one grade of expansion only.

* EQUALIZED LEAD AND CUT-OFF. †

It has been shown how, by different methods of proportioning the parts, the same mechanism can be laid out at will to give either exact equalization of the cut-off in fixed eccentric engines, or approximate equalization of the lead in shifting eccentric engines. It remains to be shown how a proportion of parts can be found which will satisfy both constructions, and thus obtain a practically constant and equal lead, an exactly equal cut-off for any chosen grade of expansion, and approximately equal cut-offs for all grades.

Referring again to Fig. 56, it appears that the fundamental principle of its construction is the diminution of the inclination to one another of the lead positions of the eccentric rod; and, referring to Fig. 31, it appears that the fundamental principle of its construction is the direct reverse of this, i.e., the increase of this inclination. It hence appears at once that the constructions of Figs. 56 and 31 are incompatible, and cannot be reconciled with one another. Comparing Figs. 57 and 32, on the contrary, it appears that in this general way they agree; but while in Fig. 57 the inclination of *hc* and *id* is reduced to actual zero, i.e., the rods are made parallel, in Fig. 32 *al* and *bk* are still inclined at an appreciable angle. Now the inclination of *al* and *bk* to one another in Fig. 32 can be varied at will by

† First published by the author in the *American Machinist* for March 14, 1889.

changing the length of the eccentric rod; and by choosing a proper length they can be made parallel, and the proportions so found will satisfy the construction for equal lead, and for the particular grade of expansion for which it should be drawn for equal cut-off likewise. For other grades it will, of course, satisfy the construction for equal cut-off only approximately. One qualification must, however, be made. In Fig. 57, A and B are located at the dead points, while in Fig. 32 they are located at the points where admission occurs, and these are not usually the dead points. It will simplify matters to make these points coincide by considering, in the first instance, an engine with a lead of zero. In Fig. 60 the construction of Fig. 32 is repeated for the mean throw of the eccentric and with lead zero, but with several lengths of eccentric-rod, giving a corresponding number of points k', k'', k''', l', l'', l'''. It is plain that the inclination to one another of $k'b$ and $l'a$ is less than that of $k''b$ and $l''a$, which in turn is less than $k'''b$ and $l'''a$, the degree of inclination depending on the length of rod used. If a length of rod can be found such that its two positions shall be parallel to one another, the rod so found will obviously satisfy the constructions for both equal lead and equal cut-off. This length is found in the following manner: It is obvious that all the points k', k'', etc., are on the straight line pg, and similarly l', l'', etc., are on qg; therefore, draw pg and qg. Assume a trial length of eccentric-rod, and with centres a, b, strike arcs giving k, l, such that bk and al are parallel, repeating the construction with different lengths of rod until the correct length is found. Locate the rocker fulcrum so

that the eccentric rod pin shall pass through *k* and *l*. Now positions *al*, *bk* obviously satisfy the construction for equal cut-off, and, being parallel, they also satisfy that for equal lead; and an engine with its valve-motion laid out in this manner will have approximately equal and constant lead, equal cut-off for that eccentric throw for which the construction is made, and an

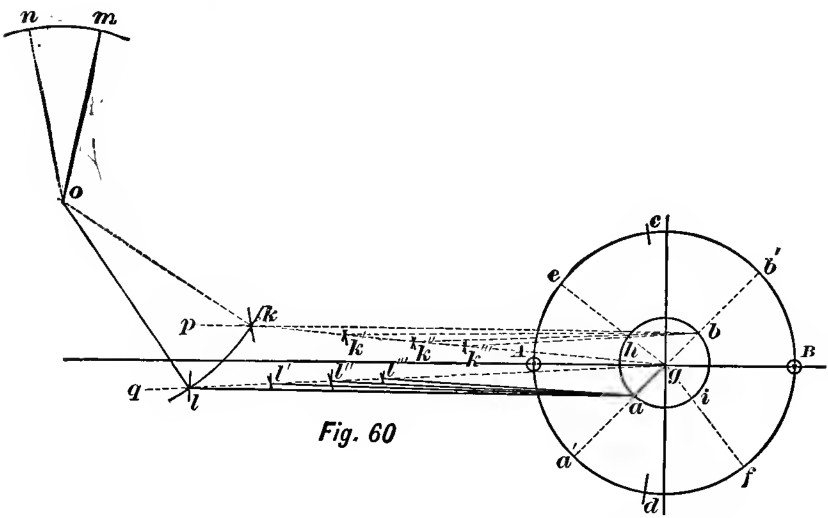

Fig. 60

approximately equal cut-off for other positions of the eccentric.

The construction of Figs. 56, 57, and 58 was made with the crank on the centre, while that for Figs. 31 and 32 was made with the crank in position for admission of steam, and in Fig. 60 these methods were reconciled by supposing the admission to occur on the centre. No material error would be introduced if this

method were followed in all cases; but if it is desired to follow the strictly correct method, it can be done in the following manner: Fig. 61 is a reproduction of Fig. 32, constructed for the mean throw of the eccentric, with some additions. Find points b_1, a_1 and k_1, l_1 cor-

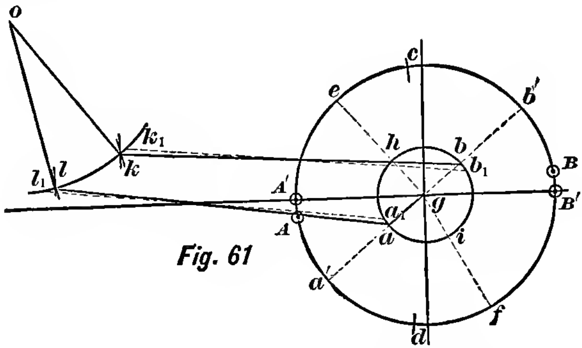

Fig. 61

responding with crank positions A', B', and draw k_1b_1 and l_1a_1. Now the construction of Fig. 32 gives the positions of kb and la, while it is desired to make k_1b_1 and l_1a_1 parallel by suitably selecting the eccentric rod length. This should be done by trial, as before, repeating the trial until the desired result is reached.

PART III.

THE SLIDE VALVE WITH INDEPENDENT CUT-OFF.

THE SLIDE VALVE WITH INDEPENDENT CUT-OFF.

INTRODUCTORY REMARKS.

AN early cut-off being a necessity for an economical use of steam, it comes about that with valves of the construction previously described the leading considerations in their design are those pertaining to the steam side of the valve. The valve and eccentric being designed with reference to the steam side, so as to secure an early cut-off, there is little that can be done with the exhaust side beyond reconciling conflicting requirements as well as possible. In engines provided with independent cut-off valves this condition no longer holds: the exhaust can usually be arranged to suit the designer's fancy; and it hence follows that in such engines the leading considerations in the design of the main valve are usually those pertaining to the exhaust. It is essential that the exhaust have a certain lead, in order that the cylinder may free itself of steam, and, on the other hand, this lead should be no greater than is necessary for this purpose, since that would involve exhausting the steam at a point where it might still do

some useful work. The determination of the point of release is, therefore, a leading factor in the design of this class of valves. It is not to be expected that there will be any close agreement in a detail of this character in the work of different designers; but, as a general rule, modified somewhat by questions of piston-speed, etc., it may be said that release should occur at from 93 to 95 per cent of the piston stroke.* The other event of the exhaust side of the valve, the compression, is determined, it must be owned, largely by the taste and fancy of the designer. A late compression by requiring a small exhaust lap conduces to a small travel of the valve, which, if it is to be unbalanced, is a desirable feature. The features of the exhaust side of the main valve and the port opening to steam having been settled, the steam side is determined by the force of circumstances. To give the proper points of exhaust opening and closure, the steam lap will usually be small and the cut-off late. This, however, is of no consequence, as the cut-off valve is introduced for the express purpose of providing for it.

THE GONZENBACH VALVE GEAR.

A description and analysis of this valve gear is here given as an introduction to those which follow. It is now seldom employed. It comprises two valve chests, two valve seats, and two valves, as shown in Fig. 62. The lower or "main valve" is driven by a fixed eccen-

* In engines of slow rotative speed—for example, Corliss engines and others of similar general character—the release will frequently be found to be somewhat later than the figure given in the text.

tric, and determines the admission, release, and compression of steam. The upper or "cut-off valve" is used solely for the purpose of the cut-off, and is usually of the gridiron type, in order to secure quickness of cut-off with moderate travel. It is driven by an eccentric of its own, which, if the expansion is to be varied, must be turned forward or backward, as the case

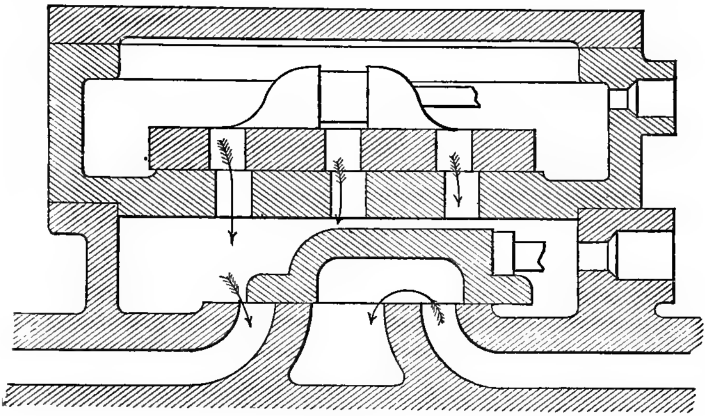

Fig. 62
The Gonzenbach Valve.

may be, on the main shaft. This movement affects the angular advance only, and, unlike the eccentric movements of Part II, does not change the travel of the valve.*

The action of this cut-off valve is different from anything that has thus far been examined. The previous

* The following applications of the Bilgram diagram to the Gonzenbach, and also to the Buckeye and Bilgram valve gears, are substantially the same as those previously published (now largely inaccessible) by Mr. Bilgram.

valves open and close their ports with the same edge, i.e., in opening the port the valve draws to one side, and in closing it resumes the previous position. This cut-off valve, however, opens and closes the port by the port in the valve passing bodily across the port in the seat, the opening being done by one edge, and the closing by the other. Further, the same ports serve for both ends of the cylinder; the valve ports passing over the seat ports in one direction for one end of the cylinder, and in the opposite direction for the other end.

It will be seen from Fig. 62 that the cut-off valve

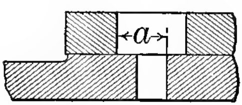

Fig. 63

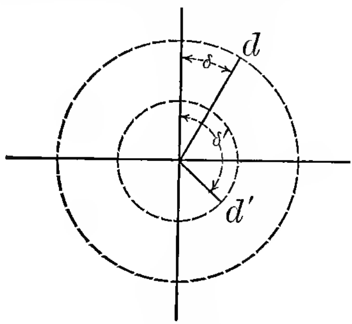

Fig. 64

has negative lap, since the ports are open when the valve stands at its central position. The width of port in the valve may equal or exceed the width of port in the seat. In the former case, the negative lap is equal to the width of port; in the latter, to the distance a of Fig. 63. The location of the eccentrics is shown in Fig. 64, d being the main and d' the cut-off; δ being the advance angle of the former, and δ' of the latter. The centres of the lap circles Q and Q', Fig. 65, are found in

the usual way, by laying off the angles δ and δ' upward from the horizontal centre line. The effect of the three ported valve being equivalent to a valve with a single port of three times the width and travel, the throw of the cut-off eccentric is in the diagram in-

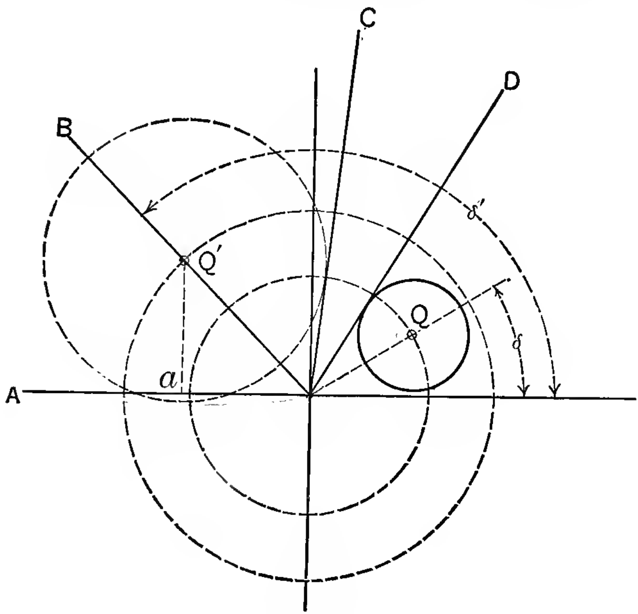

Fig. 65

creased to three times that shown in Fig. 64, and the lap circle is likewise drawn, with a radius three times the actual negative lap for each port. Starting at crank position *A*, it is clear that the main valve is open by the amount of its lead. The cut-off valve is at a distance *Q′a* from its central position, which being

less than the negative lap, the cut-off ports are open,—a fact which is also shown by the dotted (negative) lap circle going below the horizontal centre line. As the crank rises, the port opens more widely, up to position *B*. In the original discussion of the Bilgram diagram it appeared that when the crank passed through *Q* the valve stood central upon its seat. In that position the ports of the present valve stand wide open,—as is apparent from the plan of the valve and the present diagram alike. Passing crank position *B*, the valve begins to close the port, not by returning toward its former position as with previous valves, but by passing on to the other side of its centre line,—as is indicated in the present diagram, by the perpendicular from Q' falling upon the opposite side of the crank.

The closure is completed and cut-off takes place at *C*, which is indicated in the usual manner. From this point expansion goes on in the *cylinder and lower chest together*, until crank position *D* is reached, where the main valve closes its port. As has been stated, the expansion is varied by changing the angular advance of the cut-off eccentric. If δ' of Fig. 64 be increased, Q' of Fig. 65 will be lowered toward the centre line and the cut-off position *C* of the crank will be shifted to an earlier part of the stroke. This change in the cut-off will be accompanied by an earlier and earlier admission from the upper to the lower steam chest, as will be shown by the large lap circle having a larger and larger segment below the horizontal centre line. Crank position *D* extended backward gives the position at which the main valve cuts off the steam on the previous stroke, and it is clear that the cut-off eccentric might be ad-

vanced so far that the admission from the upper to the lower chest would occur before the main valve had closed its port in the previous stroke; i.e., in advancing the eccentric to obtain an early cut-off the result would be to give a second admission of steam at the latter end of the expansion. This feature limits the range of variation to the expansion which this gear can give. The only way to provide a shorter cut-off is to increase the lap of the main valve, since this carries crank position *D* backward, and allows the cut-off lap circle to be carried farther back before interfering with the main valve.

The principles of this valve, and of the application of the diagram to it, can be fixed in the mind by following the solution of

Problem VII. A Gonzenbach valve gear is to be constructed with a maximum cut-off of $\frac{5}{8}$ stroke. The greatest port opening to steam of the main valve is to be $1\frac{1}{4}''$. Since there is nothing to prevent liberality in this respect, the cut-off ports will number three, each one inch wide in valve and seat alike. Required the shortest possible cut-off, and the positions of the cut-off eccentric for the earliest and latest cut-off.

In Fig. 66 the centre *Q* of the main valve lap circle is found in the usual manner. Lead position *A* and maximum cut-off position *B* are then drawn. At the latest cut-off the cut-off valve lap circle must be tangent to *A* and *B*, and its radius being three inches (three times the port opening), it is easily drawn, giving *Q*′ the position of cut-off eccentric for latest cut-off. Extending *B* downward and finding *Q*″ such as to make the cut-off lap circle tangent to *B* extended,

determines crank position C, the earliest cut-off practicable with the dimensions given, and also the range of movement Q' to Q'' of the cut-off eccentric.

It has been shown that the range of cut-off with this gear is somewhat limited. Another defect of the arrangement is the large volume of the lower chest, which increases the clearance space during expansion. The

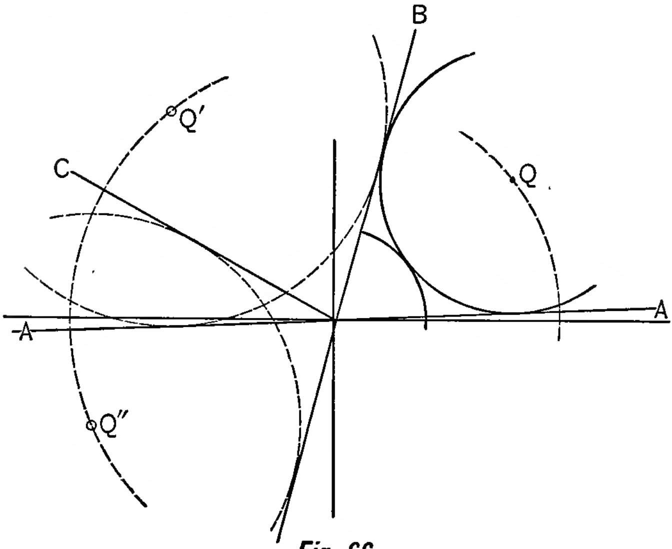

Fig. 66

object of a separate cut-off valve is to introduce early cut-off, and it is in these early cut-offs that the effect of this increased clearance is greatest, making a serious discrepancy between the "real" and "apparent" expansion. For these reasons, together with the inaccessibility of the lower valve, the plan has largely fallen into disuse.

THE MEYER VALVE GEAR.

In this gear, which has been very extensively used, a separate valve is used for the sole purpose of cutting off, as in the last example. The general arrangement of the valves is shown in Fig. 67, from which it will be seen by reference to the dotted lines that the main valve is essentially a plain valve of the usual type, with the addition of a bridge at each end to form a port through

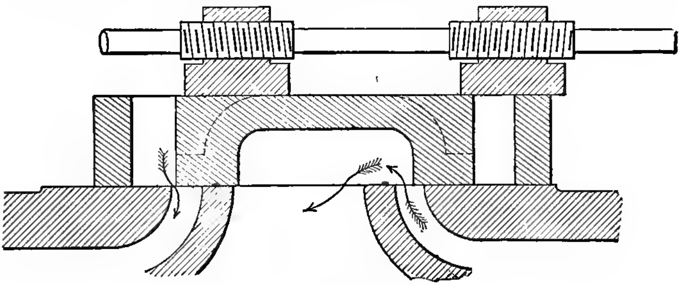

Fig. 67
The Meyer Valve.

it, and planed upon its back to form a seat for the cut-off valves. These cut-off valves are driven by a separate eccentric, and, as shown, the cut-off valve rod contains a right and a left hand screw upon which the valves are threaded. The valve rod has a hand wheel upon it, and its connection with the eccentric rod is such as to permit its being rotated at will by means of the hand wheel. This rotation increases or decreases the distance apart of the valves, and thereby changes their lap and the point of cutting off. An index is attached, which, moving over a graduated scale, shows at a glance the position of the valves upon the stem and

the degree of expansion. Occasionally this rotation of the valve rod has been connected to the governor, but the extent of the movement required and the friction incident to the mechanism are so great as to render such a plan a questionable success. Unlike the previous gear, the angular location of the cut-off eccentric is not a matter requiring exactness. In the older practice it was customary to place that eccentric exactly opposite the crank, or, since that gave the same motion, to connect the valve rod to the cross-head by means of a lever. This plan is still followed in marine, hoisting, and other engines which are to turn in both directions, since the motion of the cut-off valves is then correct for both forward and backward rotation. In present practice the cut-off eccentric of stationary engines is not usually placed so far in advance of the main eccentric. A common location is forty five degrees in advance. The effect of this is to require a smaller movement for a given change of the expansion. To offset this advantage, it reduces somewhat the width of port opening given by the cut-off valve and the speed of cutting off.

The application of the Bilgram diagram to this gear is shown in Fig. 68. The locations of the eccentrics are shown at d and d', the throw of the latter exceeding that of the former, as is customary in practice. It will be understood at the outset that since the lap of the cut-off valve is changed to vary the point of cutting off, a number of cut-off lap circles will appear in the diagram. Some of these will represent positive and some negative lap, and the determination of the proper lap for different expansions is one of the leading points to be determined from the diagram. Since the seat of

the cut-off valves is upon the back of the main valve, it is clear that the diagram must show the position of the former *in relation to the latter.* Beginning with crank

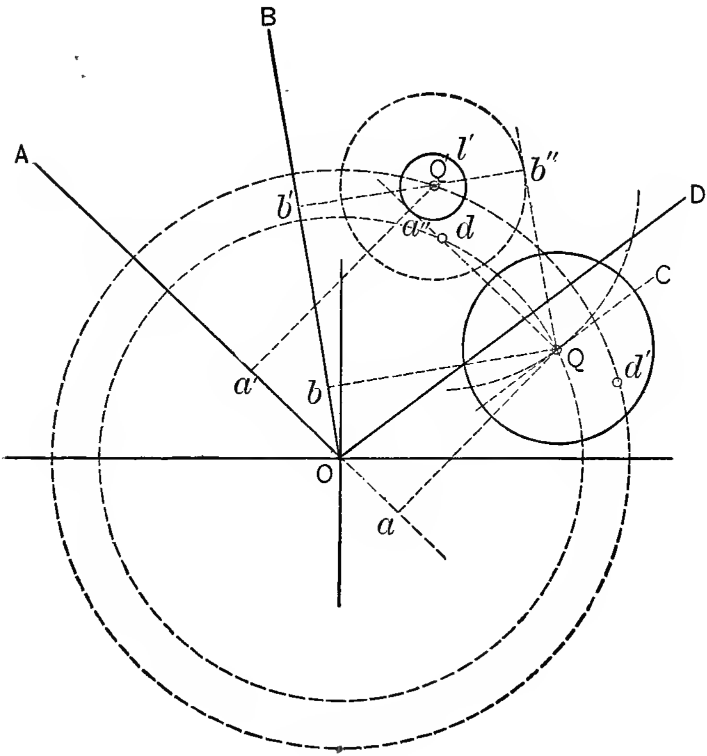

Fig. 68

position *A*, it is clear that the main valve is advanced to the right of its mid position by the distance *Qa*, and the cut-off valve by the distance *Q'a'*. The cut-off valve

is therefore removed from its mid position a distance $Q'a''$ greater than the main valve. If it is desired that the cut-off shall take place with the crank at A, there must be a lap given to the cut-off valve equal to $Q'a''$. Hence the lap circle l'. Similarly for crank position B the main valve is at a distance Qb and the cut-off $Q'b'$ from the central position. The displacement of the main valve is here the greatest by $Q'b''$, and if the cut-off valve had a lap of zero the main valve port would still be covered by the distance $Q'b''$. Therefore, if the cut-off is to occur at this position of the crank, the cut-off valve must have a negative lap equal to Q', b''. Similarly the value of the lap required for any cut-off may be found. If the range of cut off is to be from zero to that given by the main valve, the positive lap required for the former and the negative lap for the latter is easily found, and the sum of the two amounts will give the *total movement of each valve on the stem to accomplish the required range.* The tangents drawn from point Q to the various lap circles, and the perpendiculars dropped to them from the point Q', form precisely the same construction from Q as a centre that the diagram for the plain valve did from O as a centre, and these lines with the lap circles give the relations of the two valves to one another precisely as though the main were a fixed seat for the cut-off valve. As in Fig. 15, the distance Oc represented the velocity of a plain valve at the moment of cutting off, so in Fig. 68 the distances Qa'', Qb'' represent the velocity of the cut-off valve relative to the main valve at the same moment. In the figure it will be seen that this velocity is somewhat less than the corresponding velocity for the maximum cut-off by the main valve,

and in fact a somewhat sluggish cut-off is a characteristic of this valve gear. The only method of quickening this velocity is to increase the distance between Q and Q' by increasing the throw or angular advance of the cut-off eccentric. This, however, increases the diameters of the lap circles, and the distances which the valves must be moved on the stem, together with the length of steam-chest to permit the increased movement. If the full range of expansion is not required, the entire movement on the stem is available for the limited range, and this can be put to good use by increasing the quickness of the cutting off; but generally in practice it is necessary to adjust the conflicting requirements to one another with a view to securing the best compromise possible.

The greatest distance apart of the centres of the main and cut-off valves, or in other words the half travel of the cut-off relative to the main valve, is the distance QQ'. Should the cut-off plates be brought so near together that the negative lap equalled this distance QQ', the cut-off valve would close the main valve port for an instant and immediately reopen it. Should this happen before the final cut-off by the main valve, it would give a readmission of steam. To determine if this is possible, strike the negative lap circle in question with radius QQ'. Draw its tangent c through Q and a crank line D parallel to the tangent. This crank line D comes well within the main valve lap circle, indicating that the latest possible cut-off by the cut-off valve, at which the momentary closure occurs, takes place after the closure of the port by the main valve. Had the line D come tangent to the lap-circle it would have indicat-

ed that the momentary closure of the main valve port would have occurred just at the closure of the port by the main valve; and had it fallen without the lap circle, as in Fig. 69, it would have indicated that some of the later grades of expansion would have been accompa-

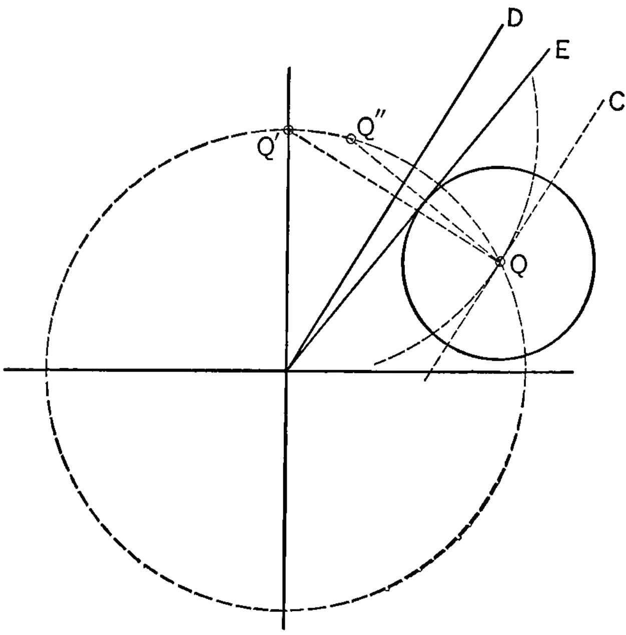

Fig. 69

nied with readmission. This is of but little moment, as, unlike the previous gear, it affects the late and not the early cut-offs; but it can be easily avoided. Thus in Fig. 69, lines *C* and *D* being parallel, *QQ′*, which is perpendicular to *C*, is also perpendicular to *D*. If the

position of Q' be altered to Q'', such that QQ'' is perpendicular to E, it is clear that D and E will coincide, the limiting condition will have been reached, and if

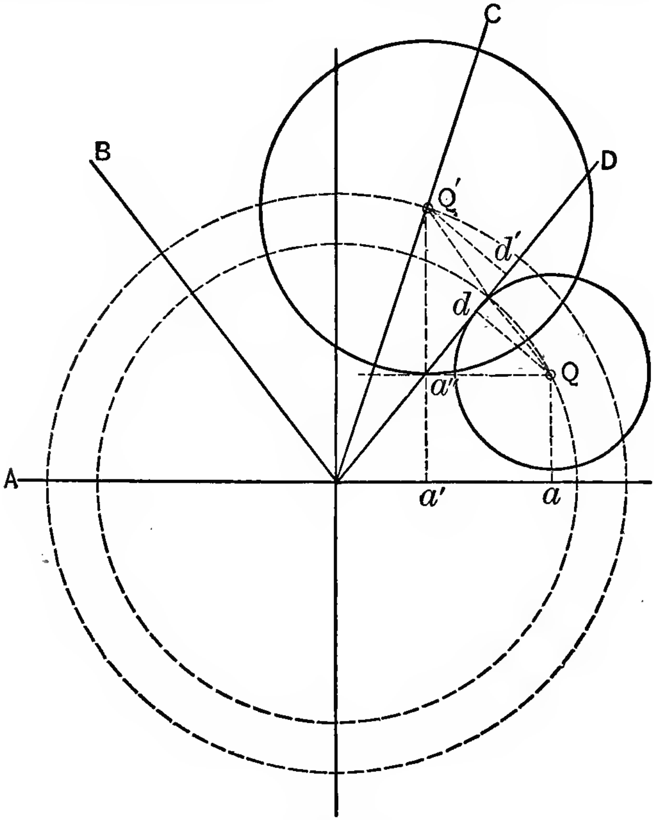

Fig. 70

the centre of the cut-off lap circles be located slightly to the right of Q'', no possible readmission can occur.

One point requiring attention remains. It is clear

that were the cut-off plates quite narrow, they might when screwed well apart pass entirely over the main valve ports and readmit steam by their back edges, and the width must plainly be made great enough to prevent this. In Fig. 70 place the crank at the dead-point position A, and determine the lap necessary for cut-off at that extreme position. The valves are located at distances Qa and $Q'a'$ from their mid positions, and $Q'a''$ is the radius of the required lap circle. As the crank mounts upward Qa lengthens and $Q'a'$ shortens, until at crank position B parallel to QQ', Qa and $Q'a'$ are equal. Beyond B, Qa exceeds $Q'a'$, until at C, $Q'a'$ vanishes. Beyond C, $Q'a'$ falls upon the rear side of the crank, and at D, $Q'a'$ becomes $Q'd'$. At this point the distance apart of the centres of the valves is $Q'd'$ plus $Q\,d$. In other words, that distance has diminished from $Q'a''$ to zero, and increased again in the opposite direction to Qa plus $Q'd'$. With the crank at A the edge of the cut-off plate just closed the port, and at D it will have closed the port by a distance $Q'a''$ plus $Q'd'$ plus Qd; and the width of the plate must equal this distance, plus the width of the main valve port, plus an allowance for tightness—say $\frac{1}{4}''$.

THE BUCKEYE VALVE GEAR.

This exceedingly ingenious valve gear is in one sense a combination of the two preceding. To the small clearance and mechanical capabilities of the Meyer valve it unites the turning eccentric of the Gonzenbach with its convenient attachment to the shaft governor.*

* The Buckeye was the pioneer shaft-governor engine.

At the same time it is not limited in range as is the Gonzenbach, and it has a sharper cut-off than either one. Balancing and other features of the valve disguise its relationships somewhat, but discussion of these features is beyond the scope of the present work. The

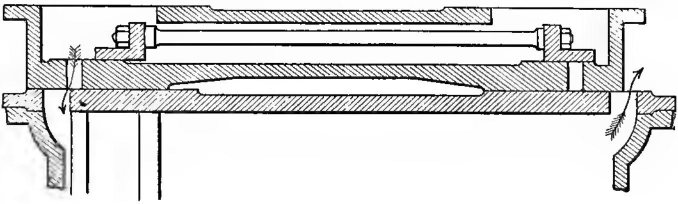

Fig. 71
The Buckeye Valve.

construction of the valve, so far as its action on the ports is concerned, is shown in Fig. 71, from which it will be seen that it takes steam from within its box-like form, and exhausts by its ends into what with other valves is commonly the steam-chest. The cut-off valves are similar to those of the Meyer system, except that they are secured immovably upon the rod.

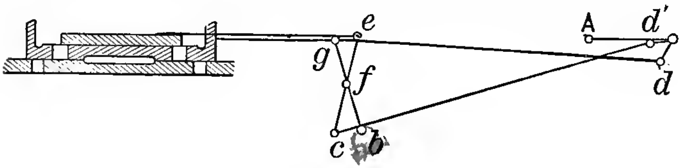

Fig. 72

Fig. 72 is an ideal view of the same valve with a diagram of the eccentrics. The position of the crank being at *A*, the main eccentric, by reason of the valve exhausting by its outside edges, is at *d*. The main eccentric and valve rods are connected to a rocker

pivoted at *b*. This rocker does not change the motion of the main eccentric in transmitting it to the valve. The cut-off eccentric and valve rods are also connected to a rocker, the former at *c* and the latter at *e*. This rocker is pivoted at *f*, which pivot is carried by the main rocker. It is clear that the motion which the cut off eccentric rod imparts to the lower end of its rocker is, if the eccentric be properly set, precisely the same as that required for a Gonzenbach cut-off valve, having its seat in line with the eccentric rod. But the distance *bc* always equals *eg*, or, in other words, the motion of the upper end of the cut-off rocker relative to the main rocker is the same as that of the lower end relative to a fixed valve seat. That is, the motion of the cut-off valve relative to the main valve is the same as that of the Gonzenbach cut-off valve relative to its fixed seat. With the Gonzenbach gear a single set of ports through the cut-off valve-seat served for both ends of the cylinder, and it was shown that, in consequence, there was danger in the early grades of expansion of a readmission of steam before final closure of the port by the main valve. With the construction of the present gear this is avoided, and, properly proportioned, there can be no readmission in either the early or late grades. The exhaust taking place at the ends of the main valve locates the eccentric diametrically opposite from its usual position, and the cut-off rocker does the same for the cut-off eccentric. The angular position of the cut-off eccentric rod also moves the cut-off eccentric from the positions shown in Fig. 73 by the same angle. Since the action of the cut-off valve is essentially the same as in the Gonzenbach gear, it may be represented

by essentially the same diagram as in Fig. 73. Q is as usual the centre of the main valve lap circles and Q', Q'', Q''' of the cut-off (negative) lap circle for different points of cut-off, Q' being the position for cut-off at

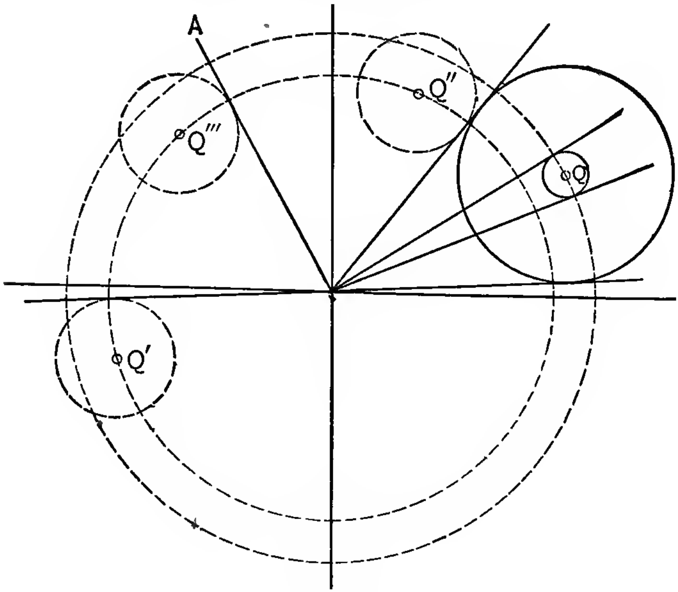

Fig. 73

zero, Q'' for latest cut-off, i.e., at the main valve closure, and Q''' for any desired crank position A. The opening of the ports by the cut off valves is in this instance of little moment, since it occurs during the previous stroke, when the admission port for the end of the cylinder under consideration is out of action.

THE STRAIGHT LINE INDEPENDENT CUT-OFF GEAR.

Fig. 74 is a horizontal section of a steam cylinder fitted with the above valves, the bottom of the figure

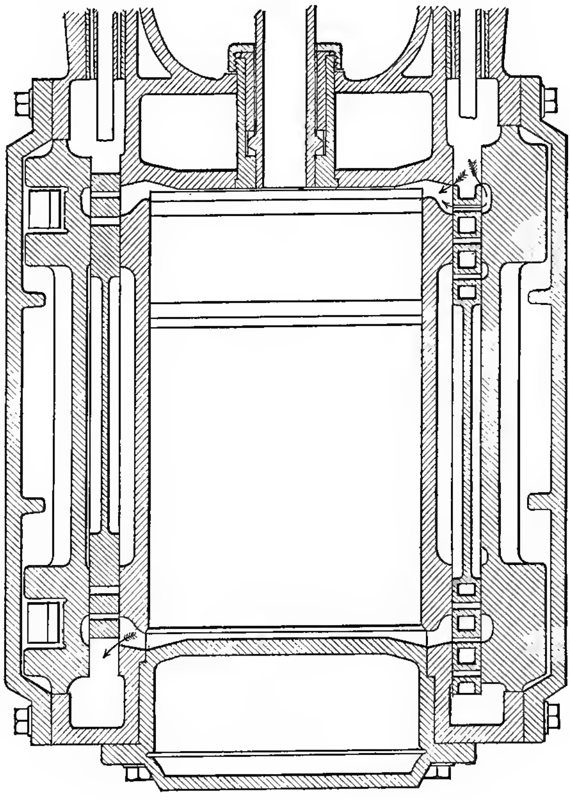

Fig. 74
The Straight Line Independent Cut-off Valve.

showing the steam and the top the exhaust valve. The two are of similar construction, both being fitted with

relief plates and multiple ports, and both acting by their outside edges. They differ chiefly in that the exhaust valve is driven by a fixed, and the steam valve by a swinging eccentric. The Bilgram diagram as applied to the gear consists of the diagram already

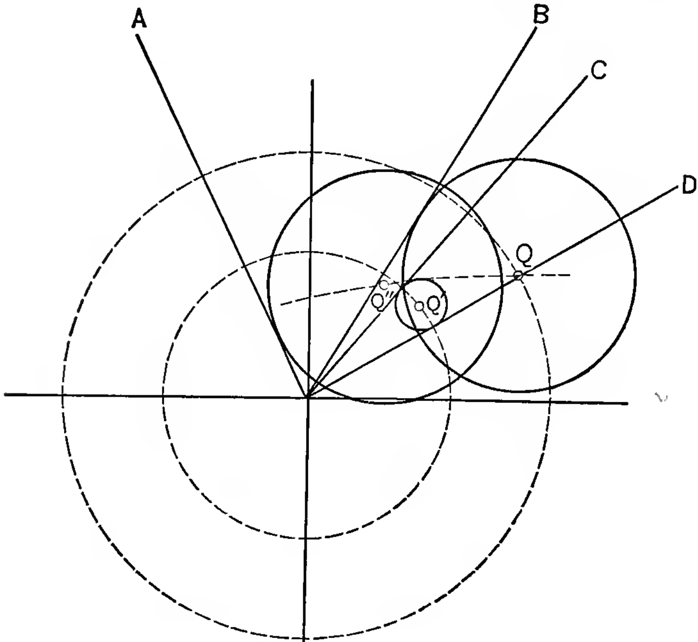

Fig. 75

familiar for the swinging eccentric gear, but with the exhaust lap circle occupying a fixed position instead of the moving one of the usual swinging eccentric gear. Contrary to all previous practice with independent valves, this engine is arranged for positive lead in the

late and negative in the early cut-offs. The object of this is as follows: It is generally understood that compression does not begin to bring the piston to rest until the pressure on the compression side exceeds that on the expansion side of the piston. With an early cut-off this state of affairs occurs at some distance from the end of the stroke, but at later cut-off the expansion curve is higher and the compression curve does not rise so soon to equal it. Hence the effect of the compression in bringing the parts to rest is lessened at late cut-off, and to make up for the deficiency a prominent lead is given.

The application of the Bilgram diagram is shown in Fig. 75, in which Q is the centre of the steam valve lap circle for greatest throw, cut-off at B; Q' is the fixed centre of the exhaust lap circle, and Q'' the centre of the steam circle with the eccentric shifted for cut-off at A, the path of the eccentric centre being QQ''. It will be observed that for cut-off at B the lead is positive and at A negative. The fixed position for compression is C, and for release D.

THE BILGRAM VALVE GEAR.

This gear has been designed to provide a more rapid cutting off than the Gonzenbach or Meyer gear. The following description is from Mr. Bilgram's book on this subject (now out of print):

Both valves are operated by one single eccentric I, Fig. 76; the main valve directly by the eccentric rod, and the cut-off valve through a peculiar mechanism, consisting of four members; viz.: the link, the rocker,

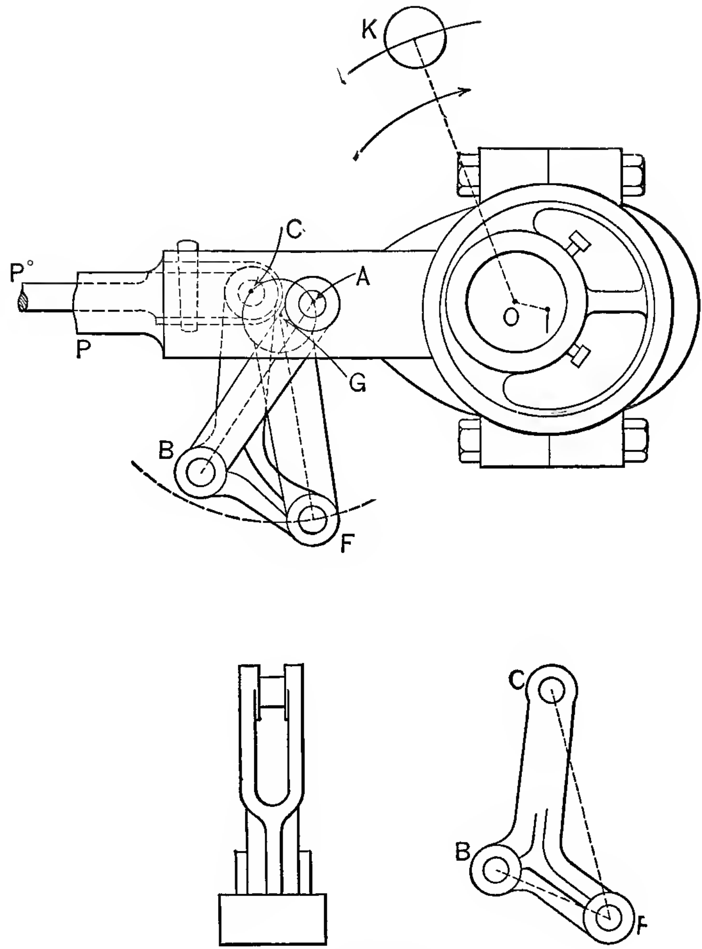

Fig. 76

the cut-off rod, and the adjustment lever. The link *AB* being jointed by the pin *A* to the eccentric rod, imparts to the rocker a rocking motion on its fulcrum *F*. The rocker is of a peculiar shape, as shown detached in the cut, but it virtually represents a bell crank (or angular lever) having an angle *BFC* of about 50°, the arm *CF* of which is about twice as long as the arm *BF*. To the extreme end *C* of this rocker is jointed the cut-off rod, by which the cut-off valve is moved. For the purpose of changing the degree of expansion the fulcrum *F* of the rocker can be moved in a circular arc, being attached to the end of the adjustment lever *GF*, whose fulcrum *G* is a fixed point.

The study of this gear will consist in the investigation of the movement of the cut-off valve for several positions of the adjustment lever. In every case we shall proceed from the neutral position of the rocker (found by transferring the eccentric rod to the centre of the crank shaft), remembering that the movement of the mechanism will be symmetrical to both sides of this position. Besides, we shall as usual neglect all complicating influences resulting from the angularity of the several members; and besides, we shall assume the movement of the pin *A* to be strictly circular and coincident with the movement of the eccentric *I*.

At first we move the adjustment lever until the line *CF* of the rocker assumes a vertical position (see Fig. 77), the theory for this position being the least complicated. When the mechanism is in operation, the pin *A* will move in a circle, and hence the points *B* and *C* will move in the arcs $b'b''$ and $c'c''$. For the latter arc we shall substitute the chord to simplify the theory.

When the crank is on its centre, the pin A will occupy the position A° corresponding to the position of the eccentric I, and since the link AB represents, as it were, the eccentric rod for the cut-off gear, we can measure the angle of advance $\delta^\circ = YAA^\circ$ by drawing AY at right angles to BA. The angle CFB being 50°

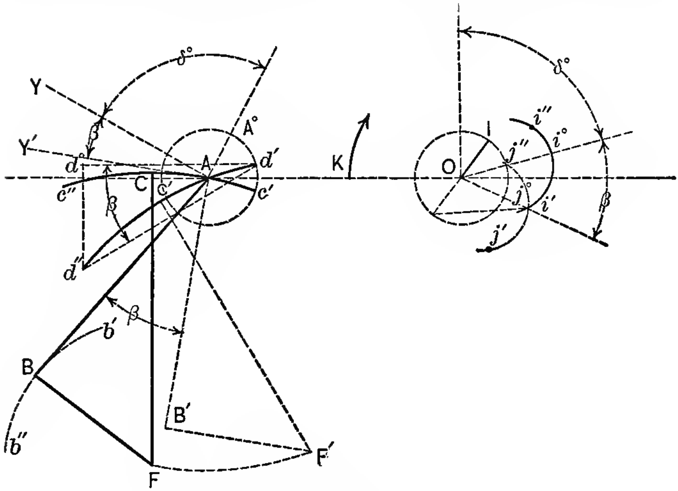

Fig. 77

and AB being at right angles with FB, or approximately so, it is evident that the angle $YAA^\circ = \delta^\circ$ exceeds the angle of advance of the eccentric I by 50°. Owing to the dimensions of the rocker, the travel of the point C, and consequently also the travel of the cut-off valve, equals double the travel of the main valve; and hence we can find the ideal eccentric i° of the movement of the cut-off valve for the considered grade by advancing the line OI through an angle of 50° and doubling its length.

Next we move the fulcrum F towards the right to F' to change the grade, and denote the angular change of the rocker by the Greek letter β. The corresponding angular change of the link AB is practically the same, and the angular advance is consequently farther increased by this angle. We can therefore draw the line Oi', but we have yet to find its length. The movement $d'd''$ of the pin C' of the rocker is doubtless the same as it was before; but being inclined, it is only its horizontal component $d'd^\circ$ that is transmitted to the valve, and the throw Oi' of the ideal eccentric for this grade will be less than Oi°. The necessary reduction can be made by drawing the line $i^\circ i'$ at right angles to Oi', which will be understood when we consider the similarity of the triangles $Oi'i^\circ$ and $d'd^\circ d''$.

In moving the fulcrum F of the rocker in the opposite direction we would have found the ideal eccentric i'', and other positions of the fulcrum F would furnish more points of the locus of ideal eccentrics. The angle $i^\circ i' O$ being a right one, it will easily be understood that all the ideal eccentrics will be located in a circle of which the line Oi° is a diameter.

These results relate to the absolute movement of the valve, and to find the ideal eccentrics for the relative movement we move the locus in the direction of and through a distance equal to IO. Having thus determined the locus $j'j^\circ j''$ of the relative movement, we can draw the valve diagram, Fig. 78, in the usual manner.

This diagram now shows that the cut-off can be adjusted to any point between the crank angles OA and OE as the angular adjustment of the rocker is not limited. It shows, moreover, that this valve gear is distin-

guished by the decided rapidity in cutting off. The closure of the steam passages is very sharp for all grades cutting-off before the half stroke; for a later cut-off however, this rapidity will get less, until when cutting off at *OE* the rapidity of the cutting off of the cut-off, valve will about equal that of the main valve.

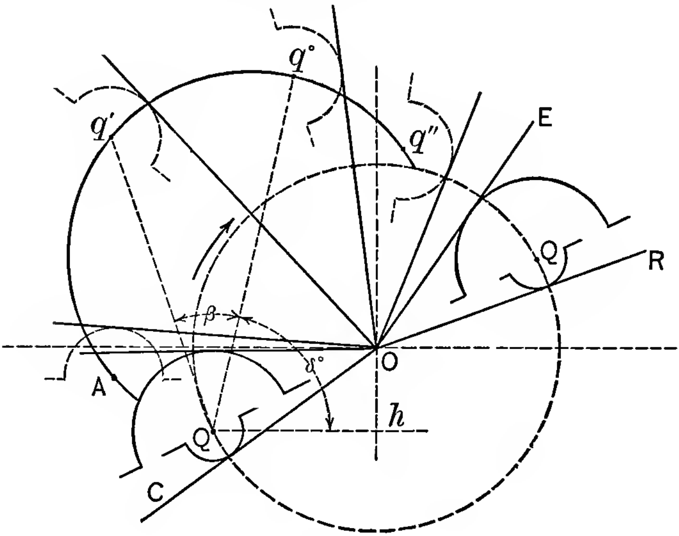

Fig. 78

The rapidity of the late cut-offs can be improved, if desired, by making the arm *CF* of the rocker more than double the length of the arm *BF*, whereby the line Oi° will be lengthened, and consequently the locus circle will be enlarged. This change entails an increase of the absolute movement of the cut-off valve above twice the travel of the main valve. Another measure consists

in increasing the negative lap of the cut-off valve; but it should never exceed the positive outside lap of the main valve. A reduction of the angle *BFC* of the rocker would likewise be efficient, but this reduction is attended by an increase of certain irregularities.

The sharpness of the cut-off will in reality be slightly less than indicated in the diagram, from the fact that the movement of the pin *A* is not circular, as was assumed, but is more or less flattened.

The proportioning of the mechanism requires some

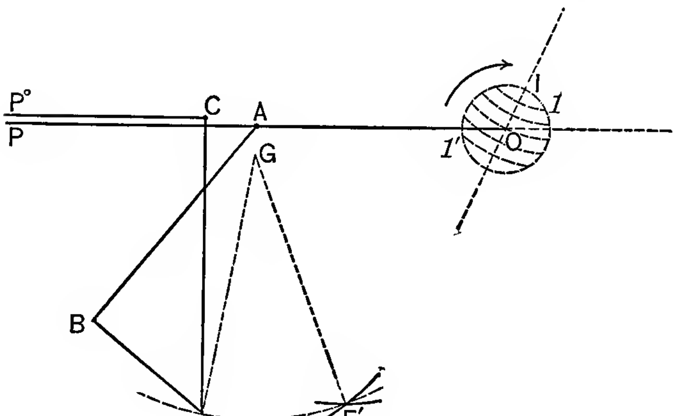

Fig. 79

care, for on it depends largely the proper operations and regularity of the cut-off. To this end we draw the rocker *BFC* (Fig. 79) with the line *CF* in a vertical position and the line *BF* at an angle of 50°, and make the arm *BF* from three to four times the throw of the eccentric, and the arm *CF* twice as long. (The figures

given have been tested by a number of experiments.) Then we draw the link AB at right angles to BF and make it about $\frac{6}{7}$ of the length of CF. The eccentric rod OP can next be shown in its neutral position, passing through the end A of the link. The cut-off rod $CP°$ can likewise be shown.

To find the length and position of the adjustment lever GF, it is necessary to make a model of thin wood or veneer, consisting of the eccentric rod, the link, the rocker, and the cut-off rod. Next we draw the orbit of the eccentric, and on it the exact position of the eccentric, say for every one sixth of the stroke of the piston, which may be done in the following way, identifying the eccentric path with the path of the crank pin:

We divide the diameter of the orbit passing through the initial position of the eccentric I in six equal parts, and draw the projection arcs of the proper radius through those points as shown. The next thing to be done is to attach the model to the drawing by joining the parts properly together with pins or thumb tacks, and fastening the ends P and $P°$ of the rods to two slides representing the valves. The right end of the eccentric rod may be provided with a needle point which at first we put in the centre O, when we set the rocker directly over the position shown in the drawing, and mark the relative position of the two slides representing the valves. Then we make two additional marks on one of them, at a distance equal to the assumed negative lap of the cut-off valve, to show the relative position for the cutting off on either side. Suppose now we desire to find the proper position of the rocker for cutting off at the point 1. To this end

we set the needle point of the eccentric rod in the point 1 of the fore stroke, and fix the valves for cutting off at the proper side, when we will find that the end F of the rocker cannot be moved but in a certain curve. This curve we mark on the drawing by setting a needle point into the rocker and making a slight scratch on the paper. Thereupon we attach the eccentric to the point 1′ of the return stroke, fix the cut-off valve to the point of closure of the other passage of the main valve, and mark another curve by the point F of the rocker. The juncture F' of the curves must of necessity be the exact position of the fulcrum of the rocker when we desire to cut off at one sixth of the stroke. In this way we can find the required position of the fulcrum for all the other grades, which will be found to form a curve. By substituting a circular arc for this curve, osculating as closely as possible, we obtain the location and length of the adjustment lever GF. An arc can generally be found to agree with the constructed curve with an almost absolute precision ; and hence it will be seen that this valve gear will admit of a practically perfect equalization of the difference between fore and return stroke.

Part IV.

THE SLIDE VALVE WITH LINK MOTION.

The Slide Valve with Link Motion.

APPLICABILITY OF THE LINK MOTION TO LOCOMOTIVE CONDITIONS.

Probably no conspicuously successful mechanical device has ever been assailed so persistently as the locomotive link motion. The steam distribution which it gives is so unlike what is considered best in stationary practice, and so like what is considered bad in that practice, that many engineers have regarded it almost with contempt as a device whose only redeeming virtue lay in the fact that it kept going. It is altogether probable, however, that these opinions are mistaken ones and that the link motion comes little short of being all that can be desired for a locomotive valve motion. The locomotive works under conditions peculiar to itself; it must in consequence be studied by itself, and conclusions based upon experience with stationary engines have little or no applicability to it. It will be shown that the action of the link motion on the movement of the valve and the distribution of the steam, is essentially the same as that of the shifting eccentric, with, however, a still more restricted port opening,

and it is to the restricted port opening, the early release and the heavy compression that the criticisms have been directed. In replying to these criticisms, the advocates of the link motion point to the well-known fact that an extreme degree of expansion in one cylinder is not conducive to economy, but the reverse, and they claim that the mean effective pressure required by a passenger locomotive at speed is so small, that if it were to be obtained by a steam distribution analogous to that given by the Corliss gear, the cut-off would be so early as to have passed the economical point, and that under such conditions, partial throttling, with its consequent superheating, is better than cut-off without throttling. The heavy compression, so far from being a defect, considered purely as a feature of the steam distribution, is pointed out to be here a positive advantage, as its effect is to reduce the cylinder capacity and thereby introduce a later cut-off than would otherwise be required. Moreover, apart from such considerations, the easy cushion due to a long compression is considered a mechanical necessity, to quietly absorb the momentum of the parts at speed, while the early release due to notching up the link is exactly adapted to the conditions, inasmuch as, when starting with the link in full gear, the speed is slow and the release is late, as it should be, while at speed, when increased exhaust lead is required to compensate for the increased speed and provide sufficient time to enable the steam to escape, it is provided by the drawing of the link toward the mid gear which accompanies increase of speed. In freight engines the mean effective pressure is greater and the cut-off later than in

passenger locomotives. The port opening is consequently larger and the compression less, so that there is little to criticise in these respects from the standpoint of the critics of the link motion, who have in fact mainly directed their criticisms to passenger locomotives. The cylinder problem of a locomotive is in fact entirely different from that of a stationary engine. With the latter, the problem is to determine the size of the cylinder and the distribution of the steam to drive economically a given load at a given speed. With locomotives, the cylinder is made of a size which will start the heaviest train which the adhesion of the locomotive will permit, and the problem then is to utilize that cylinder to the best advantage at a greatly increased speed, but under a greatly reduced mean effective pressure.

It will thus be seen that the properties of the link motion are fairly in the direction of the requirements, and while notching up the link may not give the changes desired to the exact degree which refined considerations might dictate, it does give them in the right direction and probably as nearly correctly in degree as the varied conditions would permit, even with the most elaborate valve gear.

The difficulty of the restricted port opening has no doubt been unduly magnified. There can be no doubt that many cases of loss of pressure in the cylinders of stationary engines have been attributed to the small port opening which belonged to the long steam pipes, for it is an undoubted fact that a mere diaphragm in a steam passage, which is what the port opening amounts to, does not give the obstruction to the flow of steam

that would be expected from its area. Moreover, the common rules for port opening are based on experiments with engines having a late cut-off, and there can be no doubt that with an early cut-off and the accompanying comparatively low velocity of piston up to

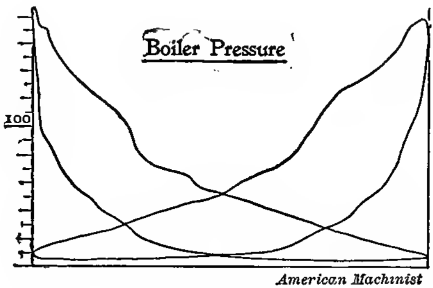

Fig. 80. Speed 55 miles per hour.

the point of cut-off, the port area required is not as large as when the cut-off is late. The indicator cards given in Fig. 80* show clearly enough that the loss in pressure at admission is much less than would be expected from the small port area and certainly show little need of a new valve gear to reduce it. Some may object that these cards are better than the average and so are not fairly representative of the work of the link motion. Be this as it may, they show what it can be made to do. Many, whose opinions are entitled to every respect, would say that they show too little throttling for the best economy. There can be no

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* By C. H. Quereau, General Foreman Motive Power, Burlington & Missouri River R. R., in Nebraska, in Proceedings Western Railway Club, March, '97.

doubt that many cases of low steam line in locomotives are due to defective mechanical features of the valve gear and not to its geometrical properties. Fig. 81* gives indicator cards which show the effects of spring in the crooked eccentric rods which are often used on consolidation and 10-wheel engines. The engine from which these cards were taken had such rods,

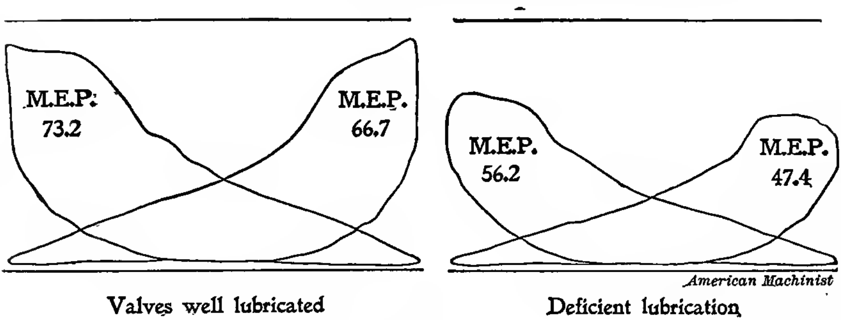

Fig. 81.

and the differences shown between the two pairs of cards were brought about by changing the lubrication of the valve—the spring of the rods being largely removed when the oil supply was sufficient.

THE STATIONARY AND SHIFTING LINK MOTIONS COMPARED.

It will be shown that the shifting link motion gives a varying lead for different rates of expansion. This has been considered by many to be a serious defect of

* By E. M. Herr, Asst.-Supt. Motive Power, Chicago & Northern Ry., in Proceedings Western Railway Club, March, '97.

this gear, and some gears have been devised largely for the express purpose of overcoming it. It is, however, probable that this supposed defect of the shifting link, like those of link motions generally, is, as a matter of fact, a point of superiority.

The variable speed under which locomotives operate must always be kept in mind in considering any feature of their steam distribution. It is certainly proper to provide a heavier cushion for high speed than for low. This all link motions do to a certain extent, by increasing the compression. In addition to this, the shifting link gives a further increase of cushion by the increase of lead. It is at least as probable that this increased lead cushion is needed as it is that it is not, while the further fact that the increased lead gives so much increased port opening at short cut-off, is certainly in favor of the shifting link, unless that link gives a larger port opening than is needed, and this its warmest advocates will scarcely claim. It is certainly difficult to see wherein an increase of lead with an increase of speed can be justly criticised.

THE LINK MOTION AND THE SHIFTING ECCENTRIC COMPARED.

The movement of a link driven by two eccentrics and hung in the usual manner is extraordinarily complex. These complexities arise from distortions of the motion which are introduced by the angular vibration of the eccentric rods and of the link itself, and by the rise and fall of the link due to its suspending stud being guided in the arc of a circle. While these distor-

tions are present, they are small in magnitude and of little real importance in the preliminary inquiry, and by ignoring them the subject may be simplified and the movement studied with a degree of accuracy sufficient for all purposes.

The effect upon the valve movement due to raising the link from the full gear position is essentially the same as that due to shifting a single eccentric across the shaft in the manner common to high-speed shaft-governor stationary engines, with the addition that by raising the link beyond the mid position the engine is reversed, while in shifting eccentric stationary engines, there being no occasion for reversal, the eccentric is not moved beyond the mid position. Were a shifting eccentric arranged to pass the mid position, such movement of the eccentric would reverse the engine and produce all the other effects of the link, and, in point of fact, locomotives have been fitted with such a valve gear.

The first step in the study of the link motion is then properly a demonstration of the similarity of its movements with those of a single shifting eccentric.

This may be done by the aid of Fig. 82*, in which, it is to be distinctly understood, the distortions already mentioned are ignored. Consequently the demonstration is only approximate, but it is sufficiently accurate for the purpose; in other words, the movements are not exactly as the demonstration would indicate though

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* Reversing the practice of the previous parts of this book, the cylinder will, in all figures of this part, be located to the right of the shaft. This is done to conform to the customary practice in locomotive drawings of showing the right hand side of the locomotive.

they are very nearly so. The dotted circle gives the path of the eccentrics, the position of the forward eccentric for the dead point position of the crank being at *a*, while that for the backward eccentric is at *b*. The shaft is supposed to have been turned forward until the forward eccentric has reached the point *c* and the backing eccentric the point *d*; the corresponding

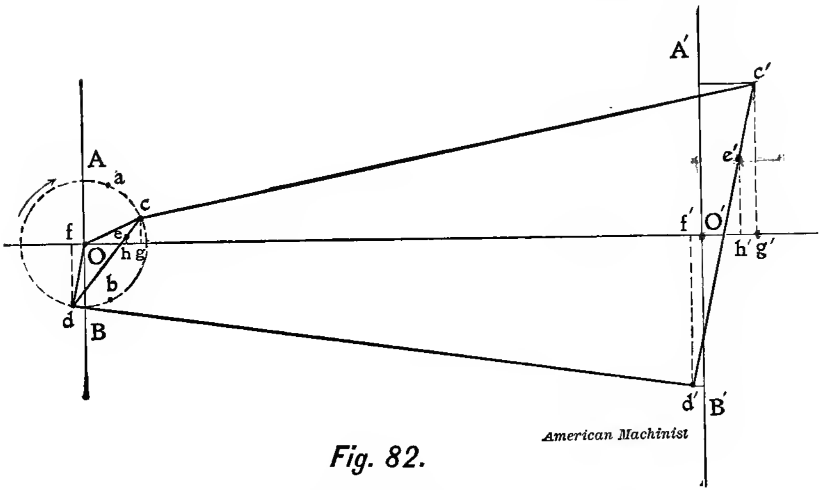

Fig. 82.

position of the link, which is here represented by a straight line, being *c′ d′*. The line *A′ B′* represents the position which the link would occupy if the centres of the eccentrics were both placed at the centre of the shaft at *O*. This line is often called the neutral position of the link, although, in point of fact, the link, as a whole, never occupies it. With the eccentrics at *a b*, the link would occupy a position parallel to this line *A′ B′*, and every point of the link oscillates equally each side of this line, but only one point of

A′B′

the link is on the line at the same time, as the link always crosses it at an angle. The horizontal movements of the points c' and d' are assumed to be exactly the same as those of the eccentric centres c and d. That is to say, it is assumed that a point dropped vertically from c' to the centre line will travel upon that line exactly as will a corresponding point dropped from c, and that a point raised from d' will travel as will one from d. It is in this assumption that the slight inaccuracy of the demonstration lies, the error of the assumption growing out of the angular inclination of the eccentric rods; but, as before stated, the error is slight and of small importance. Selecting a point as e' upon the link such that, say, $c'\ e' = \frac{c'\ d'}{4}$ and a corresponding point e such that $c\ e = \frac{c\ d}{4}$, it is required to show that the horizontal movement of e' is the same as that of e; or, in other words, that e' might be connected to e by a rod without changing the motion. It is obvious that

$$O\ g = O'\ g',$$

and similarly that

$$O\ f = O'\ f',$$

The division of the line $c'\ d'$ by e' is in the same proportion as the division of $c\ d$ by e, consequently h' divides $f'\ g'$ in the same proportion that h divides $f\ g$; that is,

$$f\ h = f'\ h',$$

subtracting $O\ f$ from $f\ h$ and its equal $O'\ f'$ from $f'\ h'$, gives us

$$O\ h = O'\ h'.$$

That is to say, the horizontal distance of c' from the central line $A'\ B'$ is the same as that of e from $A\ B$. The same proof will hold for any angle of turning of the crank, and for any position of e and e'.

THE STATIONARY LINK GIVES CONSTANT LEAD.

The above discussion relates strictly to the stationary or Gooch link in which the point of suspension is not changed in notching up, that being affected by the movement of a radius bar connected with the valve stem as shown in Fig. 83, which shows the same construction as Fig. 82, except that the link is curved convex toward the shaft, and a radius bar of a length equal to the radius of the link is added. The crank is shown in the two dead-point positions, one being given by a full line, and the other by a dotted line; the corresponding positions of the other parts being shown by similar lines. The valve is shown open to its lead for the full-line position, and it is obvious that the radius bar may be swung through the entire arc of the link without disturbing the lead; and the same would obviously be true of the dotted-line position. The feature of a constant lead is characteristic of the Gooch link, wherein it is similar to a shifting eccentric moved across the shaft in the straight line $C\ D$ of Fig. 82*.

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* In view of its slight use in this country, an examination of the action of the Gooch link, due to the fact of the paths of its two ends not passing through the centre of the shaft, by which the virtual eccentric centres are not the same as the actual, seems uncalled for.

THE SHIFTING LINK GIVES VARIABLE LEAD,

With the shifting link motion the link itself is raised or lowered to reverse the engine or shorten the cut-

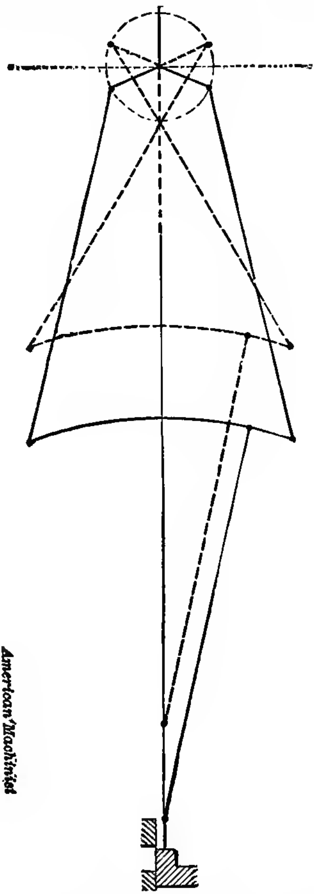

Fig. 83.

American Machinist

off. The nature of the movement of the valve is not thereby materially changed although an important dif-

ference is introduced in making the lead variable. The reason for this variation in the lead can be seen

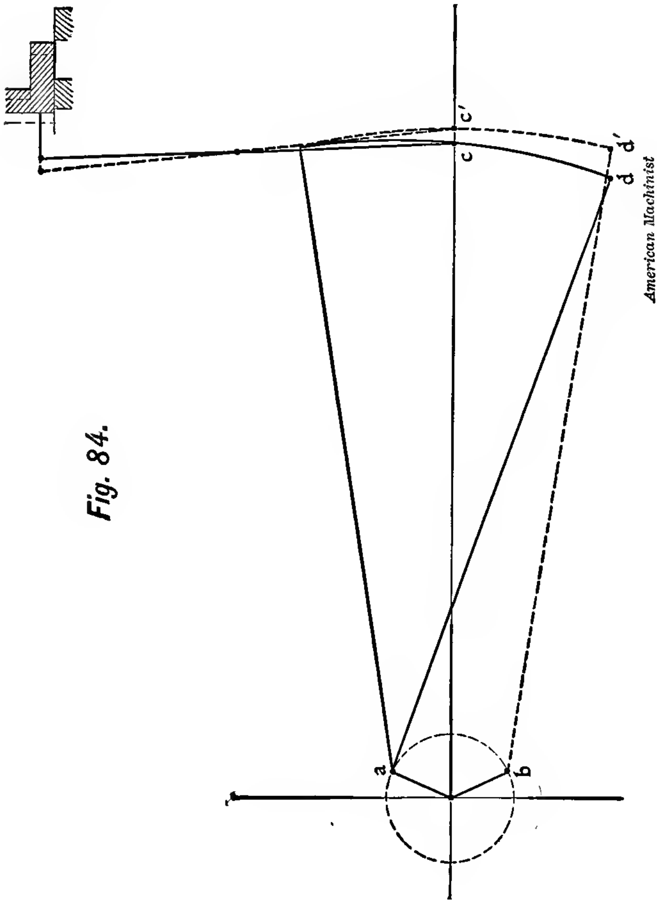

Fig. 84.

from Fig. 84, which represents the shifting link as commonly applied to American locomotives. The

crank is shown upon the forward centre, for which position of the crank the forward eccentric is at *a* and the backing eccentric at *b*, the forward eccentric rod being shown by a full line and the backing rod by a dotted line. In this position of the crank the valve has opened the port by the amount of the mid-gear lead. If now the backing eccentric be moved so that its centre coincides with that of the forward eccentric at *a*, and the link be dropped to the full gear position, the valve will take the full gear lead, and if the link arc has been struck from *a* as a centre, the link may be swung throughout its range without disturbing this full gear lead. If now the centre of the backing eccentric be brought back to its true position at *b*, the lower end of the link will be swung from *d* to *d'*, the link block will be pushed to the right from *c* to *c'* and the valve will be drawn to the left, thereby increasing the lead for the mid-gear. In other words, the full line valve sketch shows the full gear lead, while the dotted position shows the mid-gear lead.

Fig. 85 shows the same arrangements with the crank turned to the back centre, the forward eccentric now occupying the position *a* and the backing eccentric the position *b*. It is evident as before that with the backing eccentric moved to coincide with the forward eccentric at *a*, the link may be raised or lowered throughout its range without affecting the lead, but that replacing the backing eccentric to its true position, *b* will draw the lower end of the link to the left and increase the mid-gear lead as before. The proportions of these diagrams are purposely so chosen as to exaggerate this action. In actual full-sized locomotives

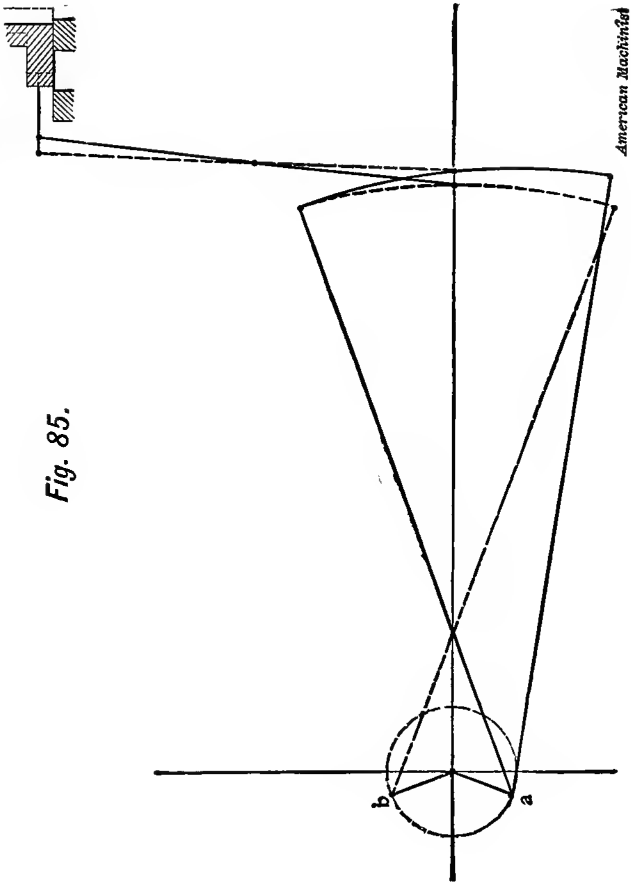

Fig. 85.

having, say, 1-16 inch in the full gear, the lead in the mid-gear is increased to from $\frac{1}{4}$ to $\frac{3}{8}$ inch.

THE CORRECT RADIUS OF THE LINK.

Figs. 84 and 85 also serve to show that the proper radius of the link is the length of the eccentric rod, plus such distance as there may be between the link arc and the eccentric rod pin—none here shown.

If the link were struck with a smaller radius using the same length of rods the effect would be an increase in the mid-gear lead in Fig. 84, but a decrease in Fig. 85, while if it were struck with a larger radius the effect would be to decrease it in Fig. 84 but to increase it in Fig. 85; that is, the effect of using any other radius than the one shown would be to make the mid-gear lead unequal for the two ends of the cylinder. It has been repeatedly suggested that the link could be curved in such a way as to compensate for the action of the rods in making the mid-gear lead greater than that for the full gear, and while it would of course be possible to find a curve for the link of very large radius such that the link block *c*, of Fig. 84, should not change its position for the forward centre as the link is raised or lowered, this would only produce a still larger mid-gear lead for the back centre as shown in Fig. 85. In other words, a link so arranged as to produce a constant lead for all grades of expansion at one end of the cylinder would be at the expense of a still greater variation for the other end of the cylinder.

LONG AND SHORT ECCENTRIC RODS.

The effect of shortening the eccentric rods is to increase the difference between the full and mid-gear

leads, as will be seen by Fig. 86, which shows the same construction as that of Fig. 84, but drawn for

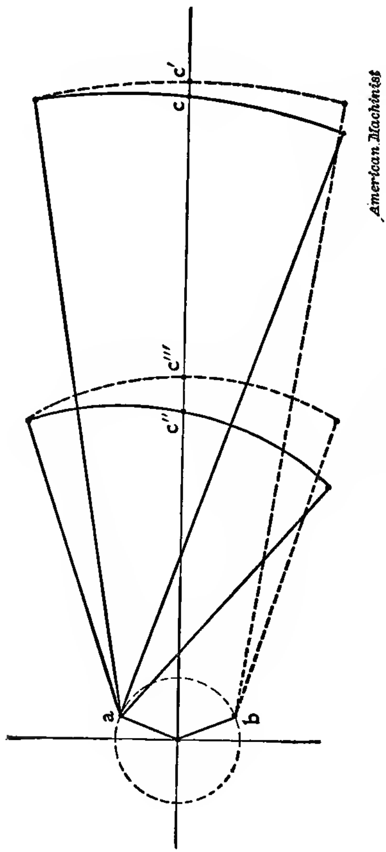

Fig. 86.

two lengths of rods. It will be seen at once that the distance between c'' and c''' is much greater than that

between c and c'. This increase in the mid-gear lead with short rods accounts for the strong effort always made to use long eccentric rods, which in the case of consolidation and ten-wheeled engines has been carried so far as to carry the rods over the forward driving axle at the expense of making them crooked. The spring of crooked rods is, however, so great as to lead to a defective valve action as has already been shown in Mr. Herr's cards given in Fig. 81, and the Schenectady Locomotive Works now make the eccentric rods in these engines much shorter, the rock shaft being placed between the forward and the next following driver, instead of forward of the forward driver. This is done because experience shows the evils due to the increased mid-gear lead of short eccentric rods, to be less than those due to the spring of the crooked rods.

OPEN AND CROSSED RODS.

A link motion arranged as shown in Figs. 84 and 85, is said to have open rods, whereas with the rods arranged as in Fig. 87 they are said to be crossed. These rods will be seen also to be crossed in Fig. 85, so that the term is not altogether fortunate. It is to be understood, however, that by the term open rods, as describing a link motion, is meant that the rods are uncrossed when the eccentric centres lie between the link and the vertical centre line of the shaft, or, in other words, that the forward eccentric connects with the upper end of the link. By diagrams similar to Figs. 84 and 85 it could be shown that the effect of crossed rods is to give a decreasing lead as the cut-off

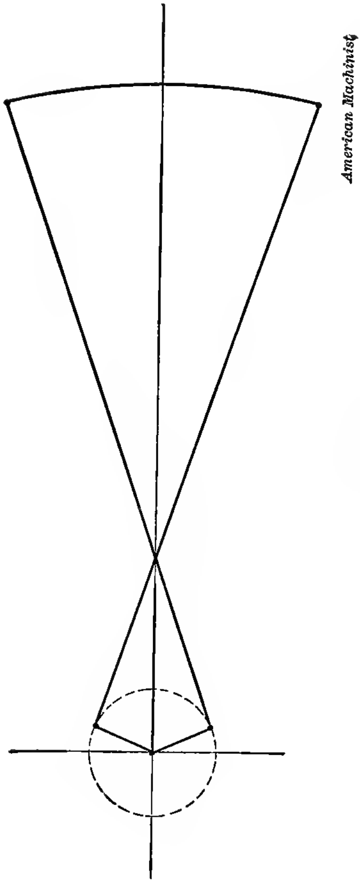

Fig. 87.

is shortened. Crossed rods are not, however, in actual use, and a detailed discussion of them seems uncalled for.

THE BILGRAM DIAGRAM.

The action of the Gooch link being thus closely analogous to that of the shifting eccentric, and the shifting link to the swinging eccentric, it follows that the Bilgram diagram as applied to these eccentric gears is also applicable to the link motion. Thus Fig. 47 represents the action of the Gooch link with constant lead, while Fig. 48 gives the action of the shifting link with open rods and increasing lead, and Fig. 49 applies to the shifting link with crossed rods and decreasing lead. In this connection, however, it should be observed that while the lead does not increase in Fig. 47, the lead angle does increase, and with it the distance through which the piston labors under the lead steam, from which it is plainly seen that a constant lead opening of the port for different cut-offs does not by any means imply a constant lead cushion to the piston, and that gears giving a constant lead do not differ so much in their effect from those giving an increasing lead, as is sometimes supposed.

Comparison of Figs. 47 and 48 will also show that at early cut-offs the port opening to steam is materially greater with increasing than with constant lead, from which deductions favorable to the shifting eccentric have already been drawn.

In stationary engines fitted with a shifting eccentric there is of course no occasion for reversal of the motion, but if desired that could be accomplished by causing the eccentric to swing past the point d^2 of Fig. 48. Such action of the eccentric in backward motion is shown in Fig. 88, the position of the crank for lead,

cut-off, release, and compression being shown for two positions of the eccentric. Should the eccentric be swung to the point d° of Fig. 48, said point d° would belong to both lower and upper arcs and being located

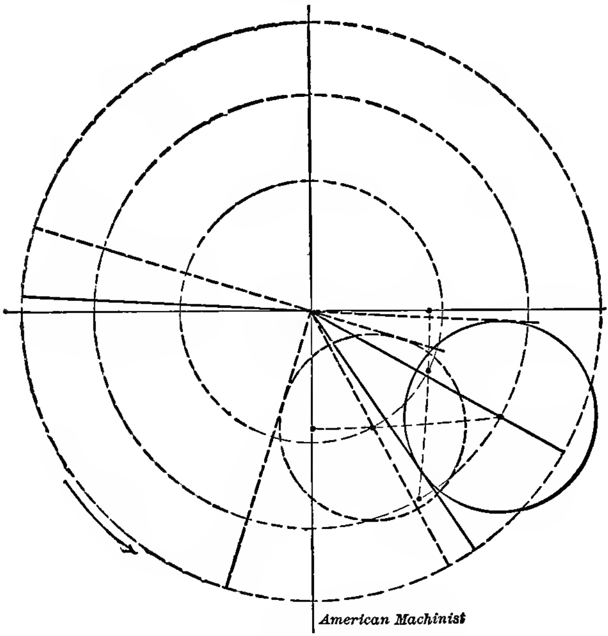

Fig. 88.

at the point where the direction of motion changes two lap circles could be drawn, as shown in Fig. 89. It will be seen in Fig. 89 that, with the eccentric in this position, the cut-off position for the forward motion is the same as the lead position for the backward

motion and vice-versa, and that the greatest port opening is equal to the lead opening; and this represents

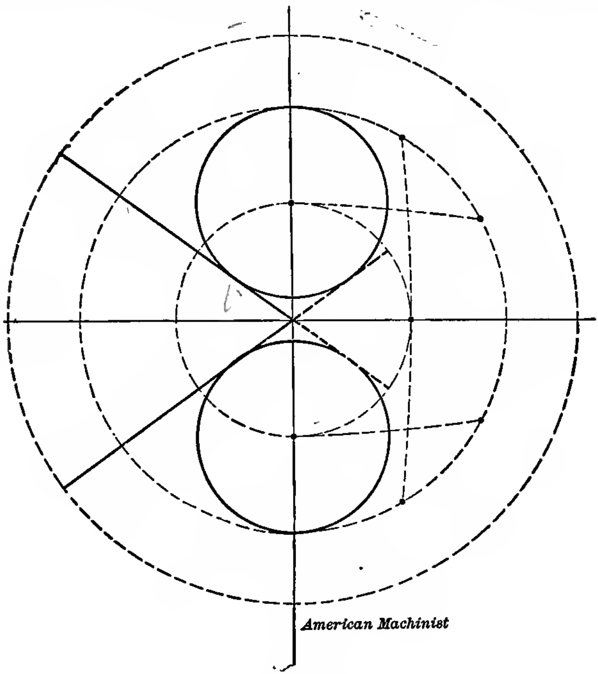

Fig. 89.

the action of an eccentric or of a link when placed in the mid-gear.

THE ACTION OF THE LINK.

The general similarity of the action of the link and shifting eccentric having been shown, it will be of service to examine the movement of the link more

minutely. In this connection it should be remembered that when the valve admits or shuts off steam it is displaced from its middle position by an amount equal to its lap. This will be apparent from Fig. 1, which is a cross-section of such a valve located centrally upon the valve seat. It will be seen that both ports are covered on the outside by the lap, and that to move the valve to the admission or cut-off position for the right-hand port involves a movement to the left equal to the lap, while to move it to the admission or cut-off position for the left-hand port, involves a similar movement to the right, and in all cases cut-off takes place with the centre of the valve approaching the centre of the seat.

In Fig. 90 *o* is the central or neutral point of the movement of the lower rocker arm and pin, at which point the valve stands centrally over its seat. This point is found in the diagram by placing the link in the mid-gear and the crank on the two centres successively, these positions being shown in Fig. 94. In these positions of the crank the valve and lower rock arm pin occupy the extreme points of their travel for the mid-gear and a point half way between the extreme points of the pin, that is half way between *a* and *b* of Fig. 94 locates *o* of Figs. 90 and 94. Measuring to the right and left of *o* a distance equal to the lap (the two rock arms being supposed to be of equal length), locates points *i* and *n*, Fig. 90, at which the ports are opened or closed, according to the direction of the movement.

The valve sketches above and to the right in Fig. 90 show the valve in these positions, the upper sketch showing the valve in the act of cutting off steam for the rear port, while the lower sketch shows a similar

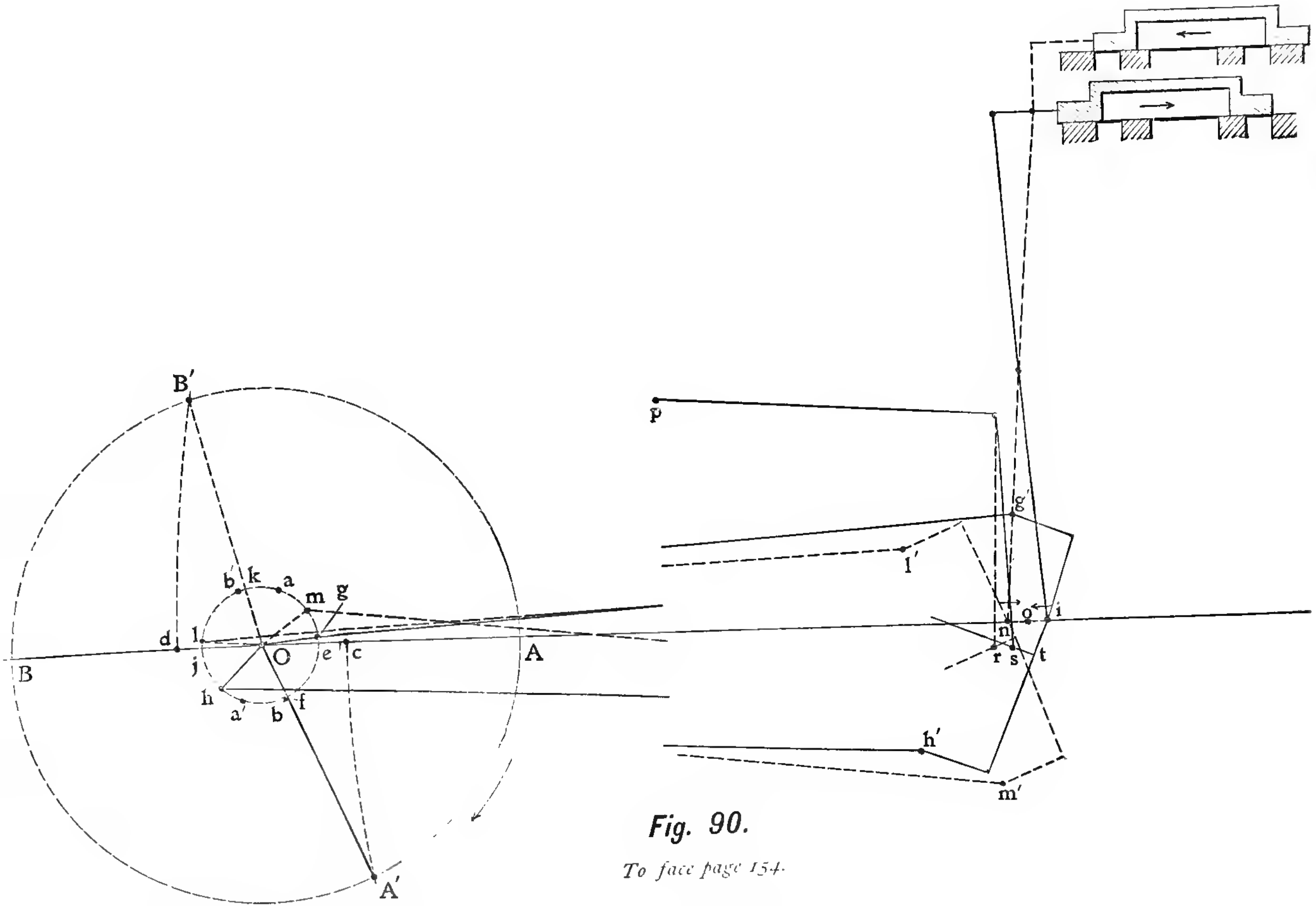

Fig. 90.

To face page 154.

action on the forward port. The sketch shows the link in skeleton diagram suspended in the usual manner by a hanger which again is suspended from the reverse shaft arm. It will be seen that this hanger is not attached to the link over the centre of the link arc, but at a considerable distance in the rear of this arc, and it will be seen at once from the diagram that the point *i* of the link at which it acts upon the link block for the forward port cut-off is farther removed from the centre *t* of the link arc than is the point *n* at which it acts upon the block for the rear port cut-off. In other words, the link is nearer the full gear position for the forward port cut-off than it is for the rear port cut-off, in consequence of which the forward port cut-off is made later and the rear port cut-off earlier than they would be if the saddle stud were placed immediately over the link arc.

The positions shown in Fig. 90 may be traced through with advantage as follows: If the crank be placed upon the forward centre $O\ A$, the forward eccentric will occupy the position a and the backing eccentric the position b, while if the crank occupies the back centre $O\ B$, the forward eccentric will be at a' and the backing eccentric at b'. If the cut-off is to be equalized at one-third stroke, points $c\ d$ may be laid down such that $A\ c$ and $B\ d$ equal one-third of the stroke. Then with a radius equal to the length of connecting rod, arcs $c\ A'$ and $d\ B'$ may be drawn giving crank position $O\ A'$, which the crank occupies at one-third stroke of the piston in the rearward motion, and $O\ B'$, which it occupies at one-third stroke of the piston in the forward motion. Taking the distance $e\ f$ in the dividers

and laying it down from a and b, point g is obtained, which the forward eccentric occupies, and point h, which the backward eccentric occupies, when the crank is at $O\ A'$. These points locate the eccentric rod pins at g', h' and give the link position shown in the full line, for cut-off at crank position $O\ A'$. Similarly by laying off $j\ k$ from a' and b', we obtain point l for the forward eccentric and m for the backing eccentric when the crank is at $O\ B'$. These points again locate the link in the dotted position for cut-off at crank position $O\ B'$. It will be seen that if the reverse shaft arm $p\ q$ be located as shown, so that the hanger swings equally each side of the vertical, points $r\ s$ will occupy a horizontal straight line, and if the reverse shaft be so located that its arm is in the horizontal position when the link is raised to the mid-gear, it will be raised as much above the horizontal line for one-third cut-off in the backing motion as it is here in the forward motion, when r and s will again occupy a horizontal line above the centre line and the cut-off will be equalized for the backward motion as the diagram shows it to be for the forward motion, and it is in this way that the reverse shaft is located. It will be seen that this method of hanging the link introduces the element of slip by which the link rises and falls on the block. Formerly it was thought desirable to reduce this slip as much as possible, and even to be satisfied with a motion which was not perfectly equalized in order to accomplish this, but at the present time constructors do not seem to be afraid of considerable slip.

ERRORS OF THE LINK MOTION.

As locomotives are built, there are three sources of error which tend to make cut-off, release and compression occur at different points in the stroke for the two ends of the cylinder. These sources of error, in the order of their importance, are, the offset of the eccentric rod pins back of the link arc, the angular vibration of the eccentric rods and the angular vibration of the connecting rod. To a certain extent the latter two compensate the first, but not entirely, and to complete the compensation the hanger stud is set back of the link arc. So far as I am aware, the importance of the first two sources of error has not before been recognized.

All previous discussions of the link motion with which I am acquainted, proceed upon the assumption that the hanger stud is adjusted to correct the irregularities due to the connecting rod, although, in point of fact, the adjustment made is the direct opposite of what would be required if this were its purpose.

If a link motion be laid down with the Scotch yoke connection between the piston rod and crank pin, which obviates the error due to the connecting rod, the offset of the saddle pin necessary to obtain an equalized cut-off will be found to be greater than if the connecting rod be introduced, and if a connecting rod be shortened, this offset will be found to diminish with each shortening of the rod, until, at some very short length, the stud will be placed over the link arc.*

* Mr. Harry Cornell, of Louisville, Ky., calls my attention to an article of his in the "Brotherhood of Locomotive Engineers'

In other words, the connecting rod, instead of being a disturbing factor, as has heretofore been taught, is in reality a corrective factor, since it, to a certain extent, corrects other errors, and in so far as it corrects these errors it reduces the offset of the saddle stud, the remaining offset being for the purpose of correcting the residual error due to the offset eccentric rod pins.

THE ERROR DUE TO THE ANGULAR VIBRATION OF THE CONNECTING ROD.

This error has already been examined at length in Part I. Considered as a source of error, in connection with link motion, it is but one of three, and in importance the smallest of the three. As given in Part I the error was stated as though it existed in the position of the cross-head—the motion of the crank pin being taken as the starting point and assumed to be true. For the study of the link motion it will be more convenient to reverse this proceedure, and to consider the motion of the cross-head as the starting point, the error being then in the position of the crank as compared with the position due to a Scotch yoke. Looked at in this way it is clear that during the rearward movement of the cross-head from *a* Fig. 91, the crank lags behind its correct position, while during its forward movement from *b*, the crank runs ahead of its correct

Journal" for October, 1896, in which he points out that the offset of the saddle stud decreases with the length of the connecting rod. It does not appear, however, that Mr. Cornell went far enough to discover the real source of the error or the real purpose of the offset

position—these errors existing at all points but being at a maximum at or near the half stroke.

The study of the effect of this action upon the link is most easily made by separating it from the other errors, and as there is a similar error due to the angularity of the eccentric rods, the study of the connecting rod error requires that the eccentric rod errors be gotten rid of by assuming the horizontal movements

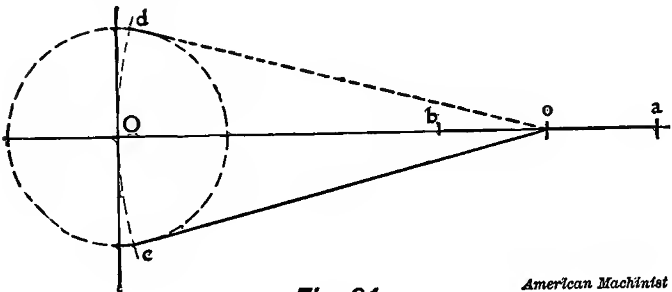

Fig. 91.

of the eccentric rod pins in the link ends to be truly the same as those of the eccentric centres. This assumption of no angular swing on the part of the eccentric rods, involves the further assumption of a straight link. Similarly, the elimination of the error due to the offset eccentric rod pins, requires the location of these pins to be on the link centre line. Fig. 92 has been made in accordance with these requirements, and shows the link positions for the crank positions of Fig. 90, with the errors due to the connecting rod included.

These errors in the position of the crank are obviously repeated in the eccentrics and link, that is for

the rearward movement of the piston, when the crank lags behind its proper position, the eccentrics and link

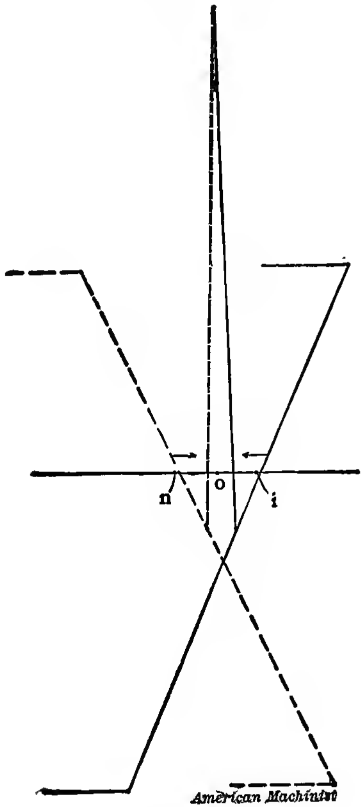

Fig. 92.

will do the same, and cut-off will not have occurred at the point desired. For the forward or return stroke

the conditions are reversed, the crank, eccentrics and link being ahead of their correct positions, and cut-off

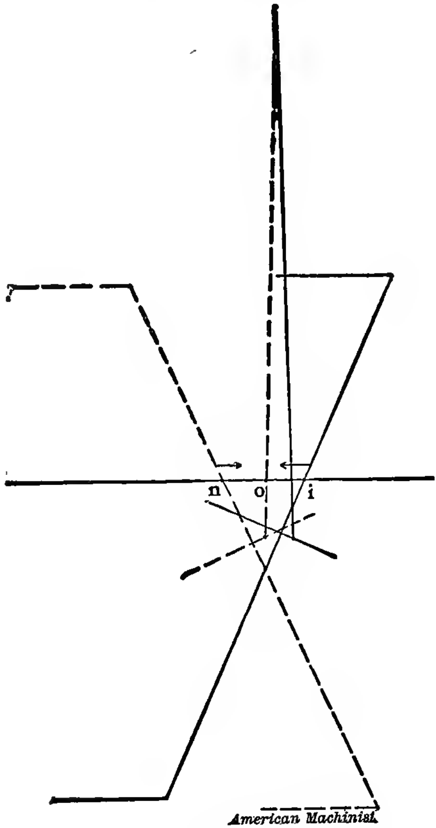

Fig. 93.

having already occurred at the desired point of the stroke. This condition of things is shown in Fig. 92,

in which the cut-off points i and n of Fig. 90 are repeated. The arrows show the direction of the motion, and it will be apparent at once that the full line link has not reached the cut-off point, while the dotted line link has passed it. It is clear that these errors could be corrected by slightly raising the full line, and dropping the dotted line position, and to do this only requires that the link stud be placed outside the link centre line, as shown in Fig. 93, and this adjustment of the stud will be seen to be the exact reverse of that actually followed in locomotives—demonstrating the position here taken, that other errors override that due to the connecting rod.

THE ERROR DUE TO THE ANGULAR VIBRATION OF THE ECCENTRIC RODS.

It is apparent at first glance that the action between the eccentrics and link ends is in a sense similar to that between the crank and cross-head. There is, however, an important difference. Reference to Fig. 91 will recall the fact that with the connecting rod the error is zero at the centres and at its maximum near the quarter, but this is not the case with the eccentric rods, because the paths of the link ends do not pass through the centre of the shaft. Referring to Fig. 94, it will be seen that the average angle between the eccentric rods and the centre line is smallest in the full line position, and that this angle increases during the entire semi-revolution and becomes a maximum at the dotted line position. In other words, the distortion, instead of increasing to a maximum for 90 degrees of

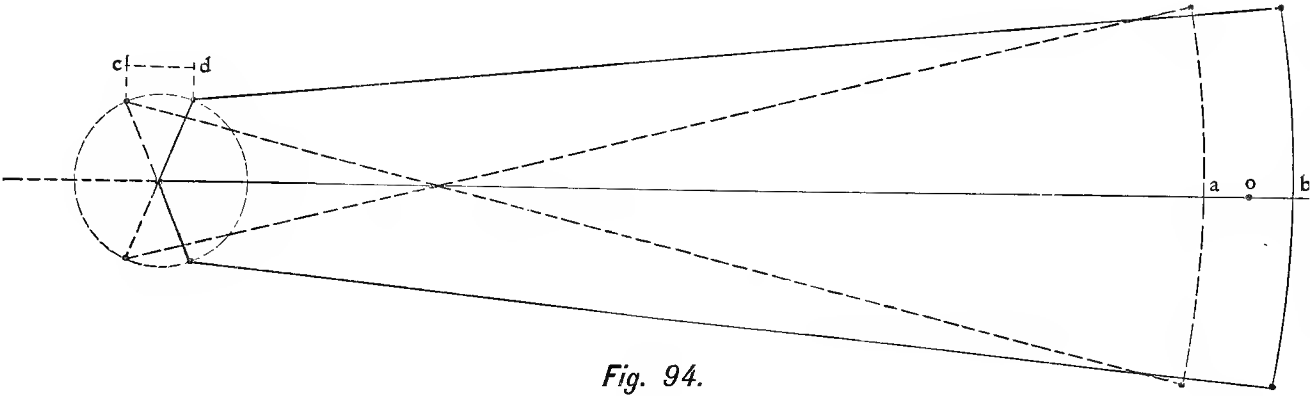

Fig. 94.

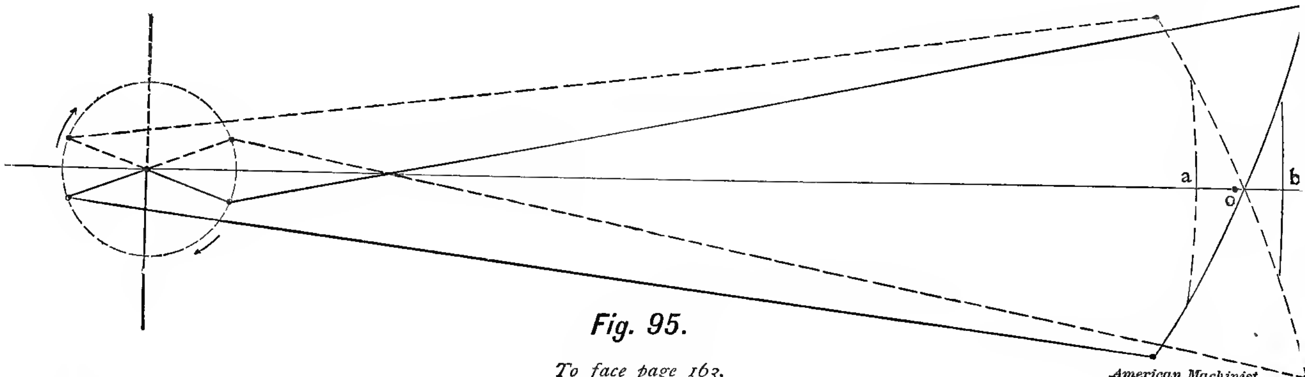

Fig. 95.

To face page 163.

rotation and then decreasing again, really increases to a maximum at 180 degrees of rotation. This increasing angle of the rods increases the movement of the link, and whereas the stroke of a cross-head is exactly twice the length of the crank, the movement of the link centre *a b* is materially more than the distance *c d* (in the case of this diagram nearly 50 per cent. more). Starting at *b*, the error in the position of the link centre steadily increases during the movement toward *a*. The errors being greatest during the second quadrant of rotation, they increase the movement more during the second quadrant than the first. In other words, the movement of the link centre is greater during the second quadrant than the first. Consequently if a diagram like Fig. 95 be laid down it will be found that the first quadrant of movement from the full line position of Fig. 94 to that of Fig. 95 leaves the link centre appreciably to the right of the neutral point *o*, and similarly a quadrant of movement from the dotted line position of Fig. 94 to that of Fig. 95 will carry the link centre to the right beyond *o*, and the same is true of any other angle of rotation. In other words, the movement of the link centre toward the left is too slow, while the movement to the right is too fast. The rocker reverses these movements on the valve, leading to too slow a movement to the right with too late a cut-off on the forward port or rearward stroke, and too fast a movement to the left with too early a cut-off on the rear port or forward stroke. These efforts are obviously in the same direction as those due to the connecting rod error, and the two are, in fact, added together in an actual locomotive. The correc-

tion of this error obviously requires an adjustment of the saddle stud in the same direction as that made to

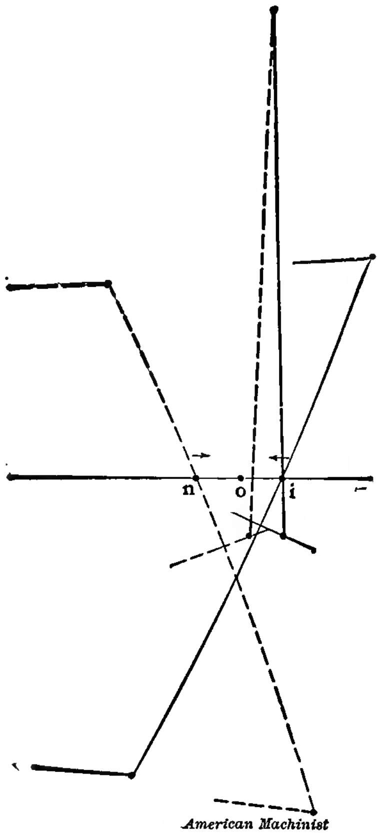

Fig. 96.

correct the connecting rod error. Fig. 96 shows this in amount, this diagram having been constructed with the connecting rod error eliminated, and the offset of

Fig. 96 will be seen to be greater than that of Fig. 93, for the connecting rod alone. Fig. 97 shows the off-

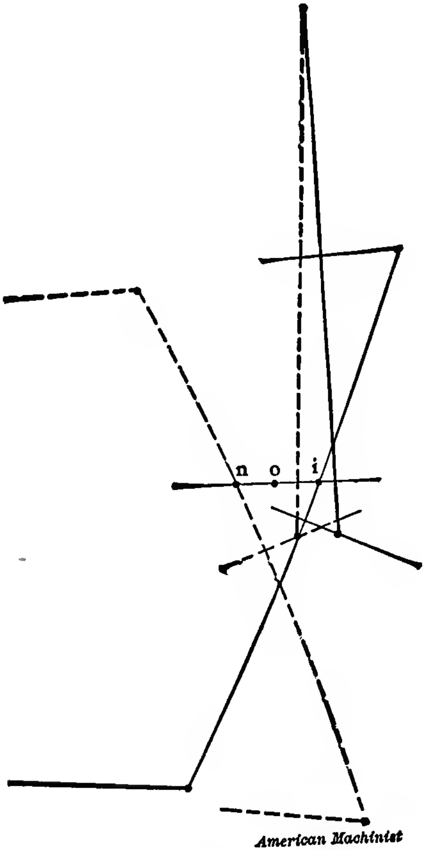

Fig. 97.

set necessary to correct the errors of both connecting and eccentric rods—the amount being approximately the sum of the offset of Figs. 93 and 96.

THE ERROR DUE TO THE LOCATION OF THE ECCENTRIC ROD PINS BACK OF THE LINK ARC.

In the previous study of the movement of the link, the eccentric rod pins were assumed to be located in the link arc, and in previous discussions of the subject it has been tacitly assumed that the errors introduced by setting these pins back of the arc are so small as to be negligible. This, however, is by no means the case, this error being, in fact, by far the most important of the three, over correcting, as it does, both the others, and resulting in finally locating the saddle stud inside the link arc, instead of outside, where the previous errors alone would place it. This error, like the others, may be best studied by isolating it so far as possible, although it is not possible to separate it from the eccentric rod error, as will be seen. It is, however, possible to separate it from the connecting-rod error. The nature of the error may be seen from Fig. 98, which shows both forms of link, one having the eccentric rod pins located in the arc, and the other having these pins located three inches back of the arc, as is customary. The eccentric rods for the former link are, of course, three inches longer than for the latter. The saddle stud is located over the centre of the link arc, as is shown in the diagram, and the links are shown approximately in the positions which they would occupy for a cut-off at one-third stroke, the full line links being in position for the rearward stroke of the piston, and the dotted line links being in position for the forward stroke. The movement of the crank

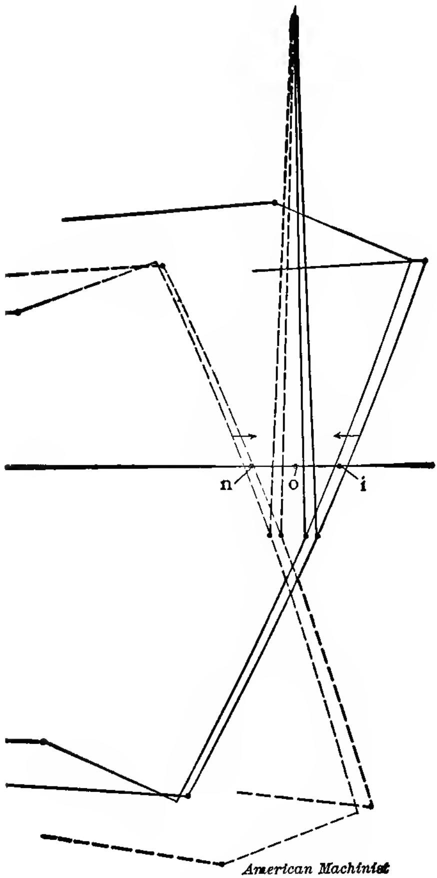

Fig. 98.

is supposed to be by a Scotch yoke, so that no connecting rod errors are introduced. It will be seen at once that the setting of the eccentric rod pins back of

the link arc makes the lines joining the extremities of the arc and the centres of the eccentric crooked, whereas with this pin located on the arc, this line is of course straight; consequently the effect of placing these pins back of the arc is for the positions shown, to draw the link having the offset pins nearer the shaft than the link which has the pins on the link arc. The action is that of a knuckle joint, any bending of which must draw the link toward the shaft. The rock shaft reverses this action on the valve, so that this drawing of the link toward the shaft pushes the valve away from the shaft. Pushing the valve away from the shaft quickens the cut-off for the front port or rearward stroke, and delays it for the rear port or forward stroke; that is, the effect of the offset pin is to make the cut-off too early in the rearward stroke and too late in the forward.

This effect will be seen to be the direct opposite of those produced by the connecting and eccentric rods, and it obviously calls for an adjustment of the saddle stud in the opposite direction, as shown in Fig. 99, which shows the position of the saddle stud necessary to equalize the cut-off at one-third stroke with the Scotch yoke connection, the stud being on the concave side of the link, where it is in all cases located in actual engines.

Summing up, Fig. 93 shows the offset of the stud necessary to compensate the error of the connecting rod alone; Fig. 96 shows the offset necessary for the eccentric rod error alone, while Fig. 99 shows the offset to compensate the effect of the location of the eccentric rod pins. The diagrams have been carefully

drawn to scale and the amounts of the offsets shows the relative importance of the three sources of error.

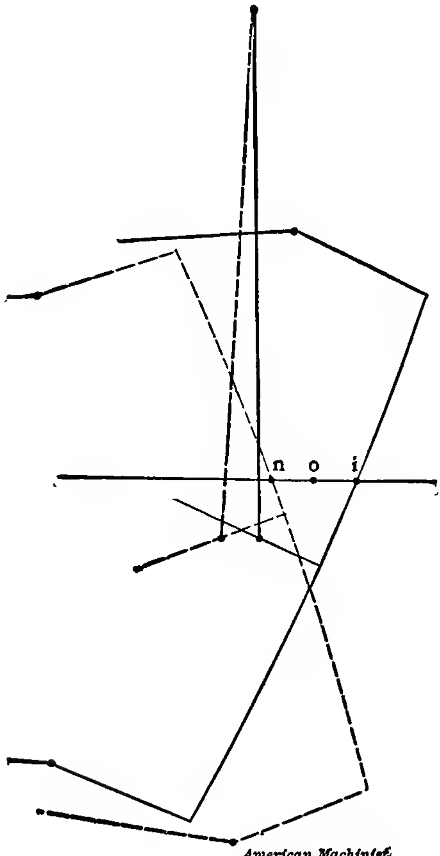

Fig. 99.

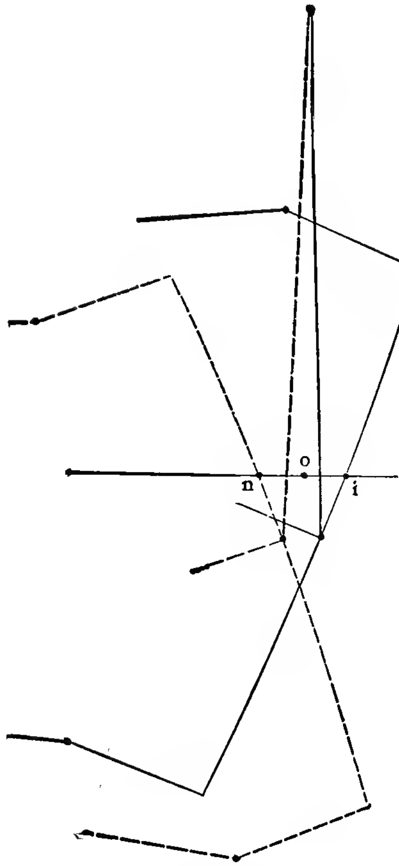

Fig. 100.

THE FINAL OFFSET.

It is obvious that the final offset of the stud is the resultant of all three. The offset of the eccentric rod

pins varies within narrow limits only, but the length of the eccentric rod varies within wide limits. It is obvious that since the error of short rods alone would place the stud farther outside the link arc than long ones, they subtract more from the offset of the stud due to the eccentric rod pins than long ones, the result being that the final offset is less with short rods than with long ones; and this is found to be the case, the offset in actual engines ranging between about five-eighths of an inch and one and a quarter inches, depending on the length of the eccentric-rods. The connecting rod also varies in length, but its influence is so small that the variation in its length between usual limits has but little effect on the final result. To illustrate this, Fig. 100 has been constructed, in which the connecting rod has been shortened by trial until the offset of the saddle stud disappeared, placing that stud over the link arc. With the other proportions unchanged, it was found necessary to shorten the connecting rod to a remaining length of three feet before this result was accomplished.

In the diagrams in which the saddle stud is located the following proportions have been used:

Stroke of piston, 24 inches.
Length of connecting rod, 91 inches.
Radius of link arc, 69 inches.
Length of link between pin centres, 12 inches.
Offset of eccentric rod pins, 3 inches.
Travel of valve, $5\frac{1}{2}$ inches.
Lap, $\frac{7}{8}$ inch.

THE ADJUSTMENT OF THE SADDLE STUD.

The adjustment of the residual error is accomplished by so suspending the link that it rises and falls in the course of its movement, in consequence of which the point acting upon the link block at cut-off in the rearward stroke is different from that acting in the forward stroke. Points of the link near the centre give earlier cut-offs than those removed from the centre, and it is obvious that by so suspending the link that the point acting upon the link block is different for the two strokes, the two points of cut-off may be altered as desired.

There are two methods by which this movement of the link can be accomplished. One, which is in universal use, consists of placing the saddle stud back of the link arc, the effect being obviously to cause the arc to rise and fall during its oscillation. The second method consists of so locating the tumbling shaft that the link hanger does not oscillate equally each side of the vertical line, but more on one side of this line than the other, the effect of which is obviously to cause the entire link to bodily rise and fall during its movement. These two methods are usually described in detail in discussions of this kind, with diagrams showing how the location of the suspension stud and of the tumbling shaft may be laid down upon the drawing board. In point of fact, however, they are not so located, and, moreover, the second method is seldom employed, as the choice of location of the reverse shaft box is usually quite restricted, and the designer is not at liberty

to place it in the position which this consideration would indicate. The location of the saddle stud is

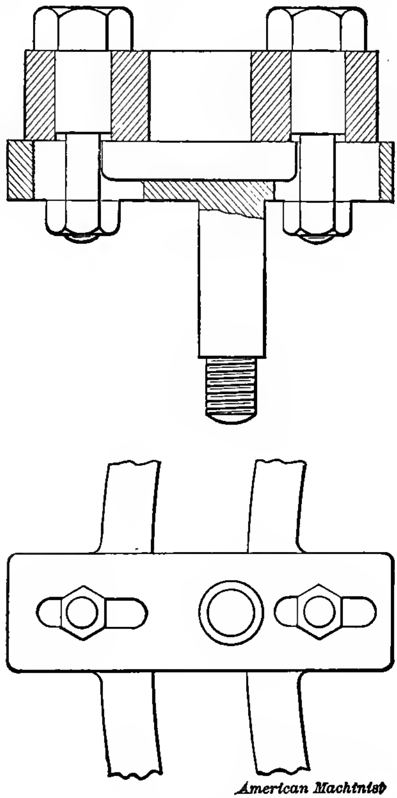

Fig. 101.

determined by trial upon the engine itself, an adjustable stud being provided, shown in Fig. 101, which is

bolted to the link, when, by trial adjustments, the proper position is found. The link is then removed from the engine and with the adjustable stud is taken to the link shop, where the permanent stud is made in accordance with it. In the case of a number of duplicate engines gotten out at the same time, the adjustable stud is applied to the first one only and the following engines of that lot have their permanent studs made in duplicate.

THE PROPORTIONS OF THE LINK MOTION.

Of design, in the sense in which that word is understood in connection with most other lines of machine work, there is very little connected with link motion. As the outgrowth of long experience, the leading dimensions of the gear are largely stereotyped in any given locomotive works, and between the work of different builders the differences are small. The following table of dimensions, taken from representative engines of various sizes as built by the Schenectady Locomotive Works, gives some of the leading dimensions. As contrasted with stationary practice, the lap for any given travel of valve is much smaller, because the cut-off in the full gear is later than would be employed in any stationary engine having a fixed cut-off, and because an increase in the lap would reduce the port opening at short cut-off. The exhaust edge of the valve varies in design between about 1-32 inch lap and $\frac{1}{8}$-inch clearance, the clearance being usually given to passenger engines.

Size and Class of Engine.	Travel of Valve.	Throw of Eccentric.	Lap.	Length of Link between Pin Centres.	Dimensions of Steam Passage.	Passage Area as a Percentage of Piston Area.	Diameter of Drivers.	Speed M. P. H.	Piston speed, feet per Minute.	Velocity of Exhaust Steam through Ports, feet per Second.	Passage Area as a Percentage of that required by rules of Stationary Engine.
	1	2	3	4	5	6	7	8	9	10	11
13x18 N. G. 8 Wheel Frt. & Pass	5	4¾	¾	10½	12x1⅛	.099	49	30	620	105	.95
14x18 Forney Suburban	5	4¾	¾	10½	12x1⅛	.088	43	25	590	112	.89
15x22—8-Wheel Frt.	5½	5¼	¾	12	14x1¼	.099	63	30	585	99	1.01
15x22—8-Wheel Pass.	5½	5¼	⅞	12	14x1¼	.099	63	35	680	115	.87
16x24—8-Wheel Frt.	5½	5¼	¾	12	14x1¼	.087	63	30	640	123	.81
16x24—8-Wheel Pass.	5½	5¼	⅞	12	14x1¼	.087	68	35	690	132	.76
17x24—8-Wheel Frt.	5½	5¼	¾	12	16x1¼	.088	63	30	640	121	.83
17x24—8-Wheel Pass.	5½	5¼	⅞	12	16x1¼	.088	68	35	690	131	.76
18x24—8-Wheel Pass.	5½	5¼	⅞	12	16x1¼	.079	67	35	705	149	.67
19x24—10-Wheel Frt.	5½	5¼	¾	12	18x1¼	.08	63	30	640	134	.75
19x24—10-Wheel Pass.	5½	5¼	⅞	12	18x1¼	.08	68	40	790	165	.61
19x24—8-Wheel Pass.	6	5¾	1¼	13	20x1⅜	.097	75	45	805	138	.72
20x24—8-Wheel Pass.	6	5¾	1⅛	13	20x1⅜	.088	73	45	830	157	.64
20x24—10-Wheel Frt.	5½	5¼	¾	12	18x1¼	.072	63	30	640	148	.68
20x24—10-Wheel Pass.	5½	5¼	⅞	12	18x1¼	.072	68	40	790	183	.55
20x26—10-Wheel Frt.	6	6	1	13	18x1⅜	.079	57	25	635	134	.75
20x26—10-Wheel Pass.	6	5¾	1⅛	13	18x1⅜	.079	68	40	855	180	.55
20x26—Consolidation slow Frt.	5½	5¼	¾	12	18x1⅜	.079	50	20	585	123	.81
21x26—Consolidation slow Frt.	5½	5¼	¾	12	18x1⅜	.072	50	20	585	135	.74

PROPORTIONS OF REPRESENTATIVE LINK MOTIONS.

From the above table it will be seen that, with a valve travel of 5 or 5½ inches, the lap in freight locomotives is uniformly ¾ inch. In the case of the 20 x 26 ten-wheeled freight engine, in which the travel is 6 inches, the lap becomes 1 inch. In the case of passenger engines having a travel of 5 inches, the lap is ¾

inch, and in case of those having a travel of $5\frac{1}{2}$ inches, $\frac{7}{8}$ inch. With passenger engines having a lap of 6 inches, the lap becomes $1\frac{1}{8}$ and $1\frac{1}{4}$ inches. The difference between the travel of the valve and the throw* of the eccentric usually grows out of the location of the point at which it is convenient to locate the rock shaft; this point being ordinarily such as to make the lower end of the rocker arm somewhat shorter than the upper end. To provide for lost motion and wear due to the numerous joints, it is customary to give the eccentric such a throw as to give a calculated travel of valve $\frac{1}{4}$ inch greater than is desired, and if the length of the rocker arms were given in this table and the valve travel calculated, it would appear as $\frac{1}{4}$ inch greater than the figures of the table. The actual travel is, however, confined to the desired amount by the limitation placed upon the throw of the reverse lever. With the exception of the 20 x 26 ten-wheeled freight engine, the length of the link between pin centres ranges between 2.2 and 2.3 times the throw of the eccentric. It will be seen that the velocity of the steam through the port is quite uniformly less in the case of freight engines than in passenger engines, owing to the fact that it is customary to use the same sized ports for a given cylinder, regardless of its duty, and the piston speed being lower in the freight engines, the velocity of the steam is less. There is sound reason in this, growing out of the fact that the terminal pressure in freight engines is higher than in passenger

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* "Throw" is here used in the locomotive sense of double the eccentricity.

engines. The usual method for calculating the velocity of the steam is, in fact, defective, as it takes no account of the increased amount of steam contained in the cylinder at high terminal pressure over that contained at low. In a general way this is met by the practice described.

THE AREA OF THE PORTS.

One of the purposes of the table on page 175 is to make some comparisons between the port areas of stationary and locomotive engines. These will be found in the sixth, tenth and eleventh columns, and to the stationary engine designer will be the most interesting part of the table. Owing to the great variations in the speed of the same locomotive under different conditions, no exact comparison of this kind is possible. The speeds desired for this purpose are the average speeds, but who shall say what these are? Under these circumstances the only basis of comparison seems to be the schedule or time-card speed, and the speeds given in miles per hour are the time-card speeds for which the engines are considered suitable, and it is from these that the piston speeds and steam velocities have been calculated. Of course the average running speeds must be greater than the time-card speeds, and the true velocities of the steam correspondingly greater, and while the port areas of the table seem small, they would seem smaller still could they be calculated from the true average running speed. As will be seen by the tenth column of the table, the velocity of the steam in locomotive ports

is, for all sizes, decidedly greater than the 100 feet per second which forms the basis of the usual rules for stationary engines. The eleventh column of the table compares the actual passages employed in locomotives with those which the stationary engineer would give, and shows them, of course, to be much smaller, even on the basis of time-card speed. Could the calculations be made on the mean running speed, there is no doubt that the ports of most of the large passenger engines would be found to have less than one-half the cross-section commonly provided in stationary engines. It should be explained that in locomotive work it is a common practice to make that portion of the passage which runs paralled with the cylinder $\frac{1}{8}$ inch wider than that leading from the valve-seat, as the inaccessibility of the first portion makes it impossible to clean it out or correct errors in the casting, while the second portion may be easily reached for that purpose. The portion of the passage, for which figures are given in the table, is the larger portion parallel with the cylinder.

PORT OPENING AND AREA OF NOZZLE.

If, however, these ports seem small, the openings of them to steam are still more so. Thus in the case of the 19 x 24 eight-wheeled passenger engine, the opening to steam at 8-inch cut-off is only 7-16 inch, and in the case of the 20 x 24 eight-wheeled passenger engine, the opening is but $\frac{3}{8}$ inch, these openings being in area but 39 per cent. and 31 per cent. of what would be required by the customary stationary engine

rule, which requires the opening for steam to be three-fourths of the port given by the rule cited above.

BLAST NOZZLE AREAS, COMPARED WITH PISTON AND PORT AREAS.

Small as these ports seem, to a stationary-engine designer, it is, however, an open question if they are not larger than necessary or advisable. The point where the steam meets with the greatest resistance in its escape from the cylinder is the blast nozzle. These nozzles were formerly made separate for each cylinder, the diameter of this pattern of nozzle for an 18 x 24 eight-wheel passenger engine ranging between $2\frac{7}{8}$ inches and $3\frac{1}{8}$ inches. Taking the mean of these, 3 inches, as the average, its area is but 2.8 per cent. of that of the piston, or but 35 per cent. of the area of the port. The port would naturally be made larger than the nozzle; but it would certainly seem that there is no occasion for so great a difference as these figures show. At present locomotives are more usually made with a single nozzle, through which the steam from both cylinders escapes, the diameters of these nozzles being about as shown in the following table, in which also their areas are compared with the combined areas of the two pistons and of the two ports. Of course a given area of nozzle arranged in this way gives less resistance than the same area in two separate nozzles, as the puff at release alternates between the cylinders; but making all allowance for this, it will be seen that the resistance of the nozzle must be far in excess of that of the port.

Size and Class of Engine.	Diameters of Single Nozzles.	Area of Mean Size Nozzle as a % of Area of 2 Pistons.	Area of Mean Size Nozzle as a % of Area of 2 Ports.
18 x 24—8-Wheel Pass.	$4\frac{1}{4}$ $4\frac{1}{2}$ $4\frac{3}{4}$	.031	.4
19 x 24—8-Wheel Pass.	$4\frac{3}{4}$ 5 $5\frac{1}{4}$	.035	.36
20 x 26—10-Wheel Frt.	$4\frac{3}{4}$ 5 $5\frac{1}{4}$	.031	.4
20 x 26—10-Wheel Pass.	5 $5\frac{1}{4}$ $5\frac{1}{2}$	.034	.44
20 x 26—Consol. Frt.	$4\frac{3}{4}$ 5 $5\frac{1}{4}$	.031	.4

SMALLER PORTS ADVOCATED.

In this connection, some remarks by Mr. Angus Sinclair, in "Locomotive Engineering" for April, 1897, are of great interest. Mr. Sinclair made use of his opportunities to measure the ports of several English locomotives, and found them to range between 10 and 13 inches in length, the width being about $1\frac{1}{4}$ inches, and Mr. Sinclair pertinently asks if the greatly increased clearance space due to the large ports of American locomotives is not at least partly responsible for the wastefulness of fuel with which they are commonly credited. In this connection Mr. Quereau's indicator cards, given in Fig. 80, are instructive. These cards were taken from an engine having its valve set to give a greatly reduced lead, as compared with ordinary practice, the result being to show clearly the termination of the compression line. Cards taken from engines having the valves set with the usual amount of lead, show the compression line running

into the lead line in such a manner as to give the impression that the compression is continuous up to or even beyond boiler pressure, and the large clearance spaces heretofore used have been considered necessary to provide room for this compression. Mr. Quereau's cards show that the compression does not reach anywhere near boiler pressure, and that the clearance volume is not needed for this purpose, and in view of this, of the above comparison of nozzle and port areas and of English practice with smaller ports, it is at least a subject of doubt if American ports are not too large for the best results.

No adequate experiments have ever been made to settle this question. It, of course, involves some risk to equip an engine with cylinders having small ports, and such experiments as have been made have been with the usual size of ports reduced at the opening by a false valve seat. Inasmuch as the expected benefits of smaller ports would be due to the reduced clearance, which is not affected by the valve seat, it is clear that no conclusions can be drawn from such experiments regarding the real value of the change. The most that can be expected is to demonstrate that the smaller ports are not harmful. It would seem that any such change should be in the width, rather than the length, of the port. A change in the length affects both admission and exhaust, while a change in width affects the exhaust only. Injury and not improvement is to be anticipated from the reduction of admission area, which should obviously be avoided.

Many years ago, Wilson Eddy, then master mechanic of the Boston & Albany Railroad, made many

engines with small ports, even down to a length of 8 inches, and his conclusions were that the standard sizes of ports were too large. In this connection, it must be remembered that back pressure is a very different factor in locomotives from what it is in stationary engines. With the latter it is an unmixed evil, but with the former it is a necessity. With stationary engines the boiler and the engine are practically independent structures, but in locomotives they are integral parts of the same machine, each reacting upon the other. The boiler not only supplies steam for the engine, but the engine, through its exhaust, supplies draft for the fire, and without the draft and the back pressure which produces it, the steam could not be made nor the engine operated.

RESULTS FROM ACTUAL ENGINES.

The adjustment of the saddle stud is, of course, made to equalize the cut-off at some one point of the stroke—usually at one-third stroke. To attempt to study the degree to which the various influences compensate one another at other points would lead us far afield, and the facts can be best seen by studying the results from actual engines. These are given in link charts Nos. 1 and 2, which are from eight-wheeled passenger engines and give a complete record of the right-hand side of two locomotives built by the Schenectady Locomotive Works. These charts are in no way exceptional, but are representative of what is done every day. The first two columns give the lead in the various gears, the second two columns give the port openings, from which the great reduction of port opening

in early gears will be seen. It should be mentioned to avoid misunderstanding, that the port opening given in the full gear means the distance by which the valve uncovers the outer edge of the port. In locomotive practice this exceeds the actual width of the port, this excess of travel being given in order to obtain more port opening in the early gears. It will be seen that the equality of the cut-offs obtained is excellent, there being no appreciable difference up to the half stroke

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LINK CHART No. I.

Schenectady Locomotive Works,
Feb. 21, 1895.

Built for St. Lawrence & Adirondack R. R. Engine No. 4437. Size of Cylinders, 18″ x 24″. Size of driving wheels, 67″.

LEAD.		VALVE OPENS.		CUT OFF.		REMARKS.
Forward Stroke.	Rearward Stroke.	Forward Stroke	Rearward Stroke.	Forward Stroke.	Rearward Stroke.	Offset $\frac{7}{8}$″
$\frac{1}{16}$″	$\frac{1}{16}$″	$1\frac{7}{8}$″	$1\frac{7}{8}$″	21″	$21\frac{5}{16}$″	Slip 1″.
$\frac{3}{32}$″	$\frac{3}{32}$″	$1\frac{9}{16}$″	$1\frac{9}{16}$″	20″	$20\frac{1}{4}$″	
$\frac{1}{8}$″	$\frac{1}{8}$″	$\frac{7}{8}$″	$\frac{7}{8}$″	18″	$18\frac{3}{16}$″	
$\frac{3}{16}$″	$\frac{3}{16}$″	$\frac{13}{16}$″	$\frac{13}{16}$″	16″	$16\frac{3}{16}$″	
$\frac{7}{32}$″	$\frac{7}{32}$″	$\frac{5}{8}$″	$\frac{5}{8}$″	14″	$14\frac{1}{16}$″	
$\frac{1}{4}$″	$\frac{1}{4}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	12″	12″	Slip $\frac{3}{4}$″.
$\frac{1}{4}$″ F.	$\frac{1}{4}$″ F.	$\frac{7}{16}$″	$\frac{7}{16}$″	10″	10″	
$\frac{9}{32}$″	$\frac{9}{32}$″	$\frac{3}{8}$″ S.	$\frac{3}{8}$″ S.	8″	8″	
$\frac{9}{32}$	$\frac{9}{32}$	$\frac{5}{16}$″	$\frac{5}{16}$″	6″	6″	

Passsenger service, four coupled wheels. Radius of of link, 69″. Length of main rods, $98\frac{1}{2}$″. Lap, $\frac{7}{8}$″.
Travel, $5\frac{1}{2}$″.

in chart No. 1, while the largest difference in the entire range is only 5-16 inch, which is found in the full gear. In chart No. 2 the action is not quite so good in the early gears, but is still better in the late ones. This surprisingly good adjustment shows that nothing of consequence is sacrificed by neglecting the location of the reverse shaft as has already been described.

Link Chart No. 2.

Schenectady Locomotive Works,
June 1, 1896.

Built for the New York, New Haven & Hartford R. R. Engine No. 4442. Size of cylinders, 20″ x 24″. Size of driving wheels, 73″.

Lead.		Valve Opens.		Cut Off.		Remarks.
Forward Stroke.	Rearward Stroke.	Forward Stroke	Rearward Stroke.	Forward Stroke.	Rearward Stroke.	
$\frac{1}{16}''$	$\frac{1}{16}''$	$1\frac{15}{16}''$	$1\frac{15}{16}''$	$20\frac{5}{16}''$	$20\frac{1}{2}''$	Slip $1\frac{1}{4}''$.
$\frac{3}{32}''$ S.	$\frac{3}{32}''$ S.	$1\frac{13}{16}''$	$1\frac{13}{16}''$	$20''$	$20\frac{3}{16}''$	
$\frac{1}{8}''$	$\frac{1}{8}''$	$1\frac{1}{4}''$	$1\frac{1}{4}''$	$18''$	$18\frac{1}{8}''$	
$\frac{5}{32}''$	$\frac{5}{32}''$	$\frac{15}{16}''$	$\frac{15}{16}''$	$16''$	$16\frac{1}{16}''$	
$\frac{3}{16}''$ S.	$\frac{3}{16}''$ S.	$\frac{11}{16}''$	$\frac{11}{16}''$	$14''$	$14''$	
$\frac{3}{16}''$ F.	$\frac{3}{16}''$ F.	$\frac{9}{16}''$	$\frac{9}{16}''$	$12''$	$11\frac{13}{16}''$	Slip $\frac{13}{16}''$.
$\frac{7}{32}''$ S.	$\frac{7}{32}''$ S.	$\frac{15}{32}''$ S.	$\frac{15}{32}''$ S.	$10''$	$9\frac{7}{8}''$	
$\frac{7}{32}''$	$\frac{7}{32}''$	$\frac{3}{8}''$	$\frac{3}{8}''$	$8''$	$7\frac{15}{16}''$	
$\frac{7}{32}''$ F.	$\frac{7}{32}''$ F.	$\frac{5}{16}''$	$\frac{5}{16}''$	$6''$	$6''$	

Passenger service, four coupled wheels. Radius of link, $60\frac{1}{2}''$. Length of main rods, $94''$. Lap, $1\frac{1}{8}''$. Travel, $6''$.

RECENT PRACTICE IN VALVE SETTING.

It was formerly the universal, as it is still the usual, practice when setting locomotive valves, to give them a small (about 1-16 inch) lead in the full gear, and then allow them to take such leads for shorter cut-offs as the proportion of the parts, especially the length of the eccentric rods, should determine. Of late a more rational method has been practised to some extent, which is undoubtedly growing and destined to grow still more. This method consists in adjusting the lead for the running cut-off, instead of the full gear, the leads for the other gears being largely determined by the mechanism. This involves a negative lead or blind port in one or both full gears; but as might have been expected, this is found to have no deleterious effect on the running qualities in the full gear, owing to that gear being used only at starting, when the speed is low, and with the large port opening of the full gear no lead is required to properly fill the cylinder with steam. In point of fact, negative lead in the full gear is claimed by many to be an advantage and to give an appreciable increase to the power of the engine when running in the full gear, due to the fact that positive lead offers resistance to the motion of the piston. The effect of the reduced lead in the running gear is to cause the engine to ride easier, to reduce the frequency of hot bearings, and, by reducing the strain to lessen repairs. A slight gain in fuel economy is also claimed by some, due to the reduction of hurtful resistance to the motion of the

piston and to a slight prolongation of expansion before release.

A leader in this movement is Mr. C. H. Quereau, of the Chicago & Missouri River Railroad, in Nebraska, who has done a good deal of experimenting on the subject and has presented his conclusions in a paper before the Western Railway Club. Mr. Quereau gives gives several indicator cards from the same engine with various settings of the lead, of which Figs, 102, 103,

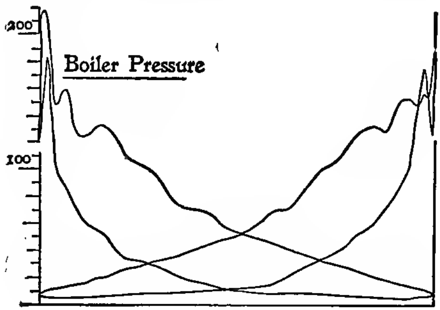

Fig. 102.

Lead at Cut-off $\frac{11}{32}$″.

SPEED 55 M. P. H.

and 104, are presented as examples. The effect of the change upon the steam distribution is apparent at once. The fact that the compression does not rise to boiler pressure, which has been already referred to, will be seen, and from which deductions regarding the size of ports has been drawn.

This practice has been adopted by many progressive superintendents of motive power and railroad master mechanics. It is endorsed by the mechanical officers

of the New York, New Haven & Hartford; Maine Central; Chicago, Burlington & Quincy; Chicago, Burlington & Northern; Chicago & Northwestern;

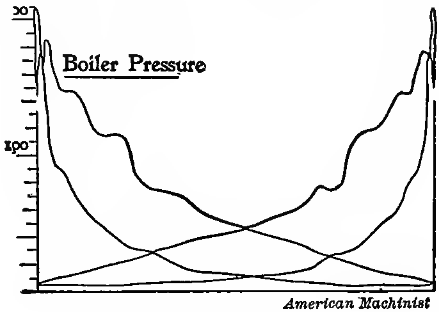

Fig. 103.

Lead at Cut-off $\frac{9}{32}$".

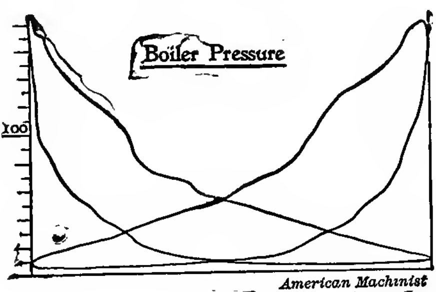

Fig. 104.

Lead at Cut-off $\frac{7}{32}$".

SPEED 55 M. P. H.

Illinois Central; Boston & Maine; Lake Shore & Michigan Southern and other leading roads.

The reduction of the lead in the running gear

may be brought about by setting back the forward eccentric or the backing eccentric, or both, and in the latter case in equal or unequal amounts. Railroad mechanics who agree on the main point of a reduced running-gear lead, differ in the method used to accomplish it. Following is the practice of several roads in this respect:

The New York, New Haven & Hartford gives to fast passenger engines 1-16-inch positive lead in full gear forward, and 1-4-inch negative lead in the full gear back, resulting in 1-4-inch positive in the running gear.

The Maine Central sets the valve line and line for the full gear forward, and gives 1-4-inch negative lead in the full gear back on passenger engines.

The Illinois Central gives to passenger engines 1-32-inch positive lead in the full gear forward and then adjusts the backing eccentric to give about 3-16-inch lead in the running cut-off.

The Lake Shore and Michigan Southern gives to passenger engines 1-16-inch negative lead in the full gear forward and 9-64-inch negative lead in the full gear back, resulting in about 5-16-inch lead for the running cut-off.

The Chicago Great Western sets the valves of passenger engines line and line in both forward and back full gears, resulting in from 3-16-inch to 9-32-inch lead at 6-inch cut-off, and on mogul freight engines gives 3-64-inch negative lead in both full gears, resulting in 1-4-inch lead at 6-inch cut-off.

The Chicago & Northwestern recently specified for some 19 x 24 passenger engines 3-16-inch negative lead in the full gear forward and 1-4-inch positive lead

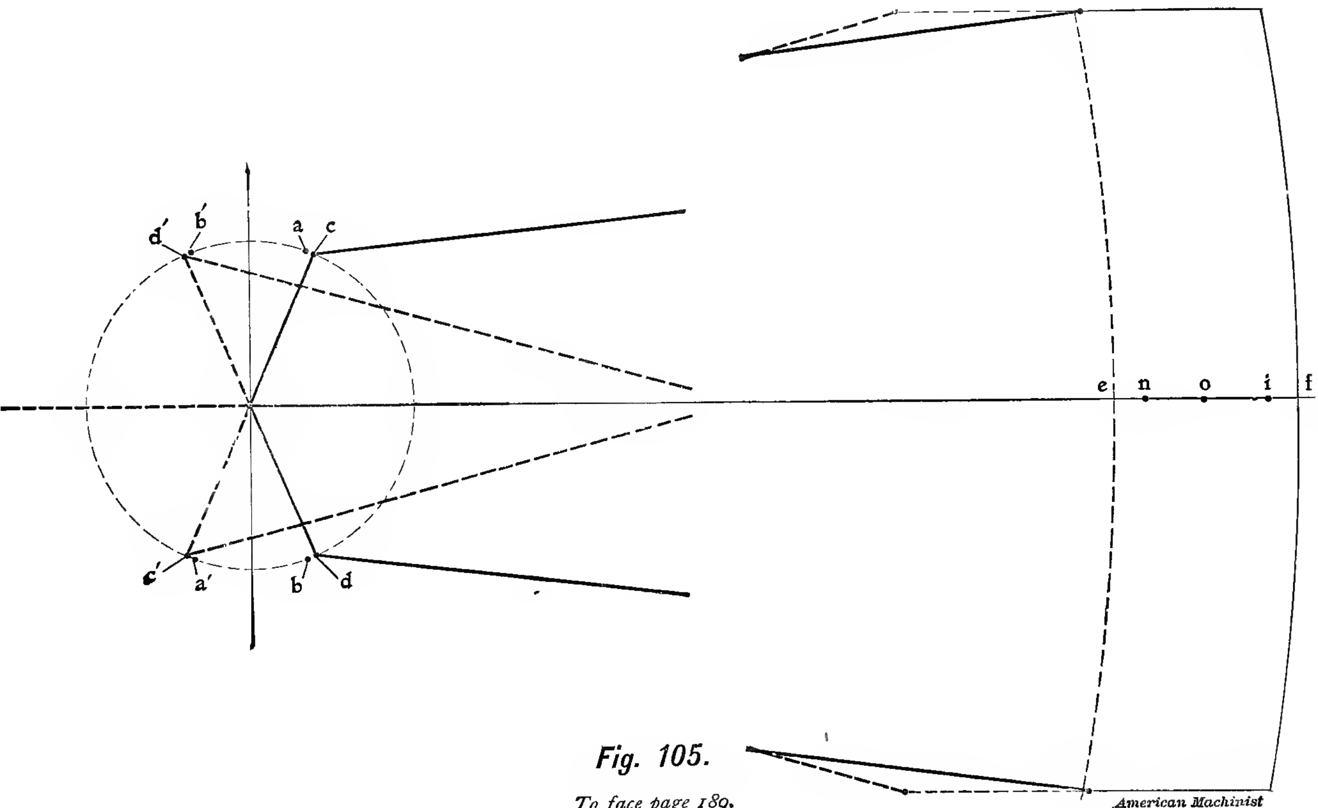

Fig. 105.

To face page 189.

at 6-inch cut-off, obtained by adjusting the backing eccentrics. These engines had Allen valves.

The diversity of practice is apparent, but doubtless many of the apparent discrepancies would dissappear if the full details of the motions were known. The effect of the length of the eccentric rods is so marked on the midgear leads that it has doubtless had an influence on the methods followed in different cases, even when substantially the same running gear lead has been arrived at.

The analysis of the action of the various settings of the eccentrics can best be made by means of the Bilgram diagram. Referring to Fig. 48, it will be seen that in order to determine the properties of the gear it is only necessary to know the location of the line QQ°, and on this draw the lap circle for various points of cut-off. The point Q may be located directly from the lap and the full gear lead, and the point Q° from the lap and midgear lead. The full gear lead is usually assumed at the beginning, but the midgear lead varies with the length of the eccentric rods and must be found. This is done in the manner shown in Fig. 105. The diameter of the dotted circle is equal to the full gear travel of the valve—6 inches—and points $a\ b\ a'\ b'$ are laid down at a horizontal distance from the vertical centre line equal to the lap $1\frac{1}{8}$ inches, From these points the full gear lead 1-16-inch is laid down, giving points $c\ d\ c'\ d'$, which the eccentrics occupy when the crank is on the centres. From these points the full and dotted positions of the link are easily found, with the resulting midgear travel $e\ f$, which by measuring is found to be $3\frac{3}{8}$ inches. Dividing this in half and sub-

tracting the lap *o i* or *o n* gives the midgear lead *i f* or *e n*, which is equal to $\frac{3\frac{3}{8}}{2} - 1\frac{1}{8} = \frac{9}{16}$ inch. One

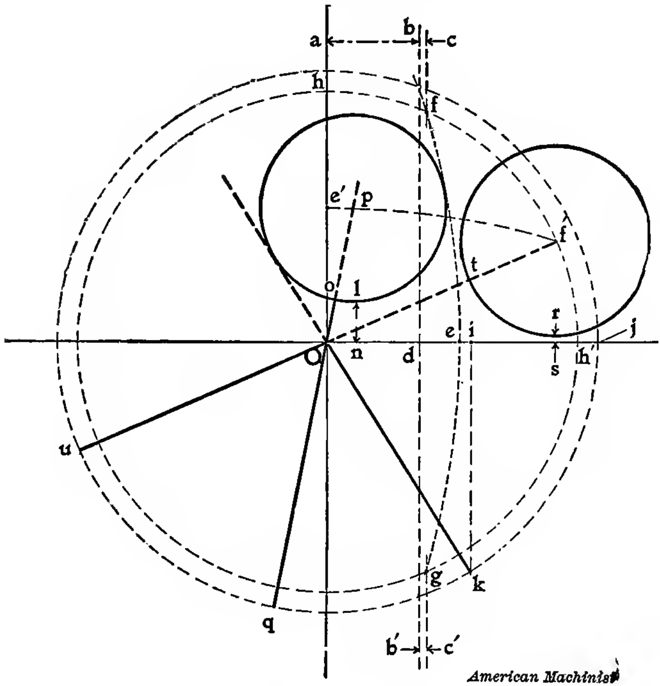

Fig. 106.

ab = lap.
$bc = rs$ = full-gear forward lead—positive.
$b'c'$ = full gear backward lead—positive.
$de = ln$ = mid-gear lead.

caution should be given here. When the eccentric throw and valve travel differ, it is most convenient to use the valve travel as the diameter of the circle, but

when that is done the eccentric rod lengths and the link dimensions should be enlarged from the actual in the proportion of the valve travel to the eccentric throw.

Having the midgear lead determined, the Bilgram diagram, Fig. 106, can be constructed thus: draw the inner dotted circle equal in diameter to the full stroke valve travel, and lay down *a b* equal to the lap, make *b c* equal to the full gear lead, and *d e* equal to the midgear lead, as found in Fig. 105. Now, a circle struck through *e f g* will give the path on which a single shifting eccentric must travel to produce a valve movement equivalent to that given by the link of Fig. 105. Laying off $h' f'$ equal to *h f*, and $O e'$ equal to *O e*, the circular arc $e' f'$ is easily drawn, on which the centre of the lap circle for all points of cut-off must lie. Drawing the outer dotted circle to represent the path of the crank pin to scale, and selecting, say, the cut-off at one-third stroke for study, the point *i* is laid down such that *i j* equals one-third of the stroke, and by the perpendicular *i k* the crank line *O k* for one-third stroke is located. This is extended upward,* and the lap circle is drawn tangent to it with its centre on the line $e' f'$. We now have for the one-third cut-off a lead equal to *l n*, a port opening equal to *O o*, a travel equal to twice *O p*, an exhaust opening and closure (assuming no inside lap) at crank position *O q*. Simi-

* Because the American locomotive valve gear always has a rocker. It will be recalled that the previous Bilgram diagrams were for gears having no rocker, in which case the lap circle is tangent to the crank centre line and not to its extension.

larly we have for the full gear a lead *r s* equal to *b c*, a port opening *O t*, a travel equal to twice *O f'*, and an exhaust opening and closure at crank position *O* ~~*x*~~. *u*

The effect of making the full gear leads negative is seen in Figs. 107 and 108. Proceeding in Fig. 107 as before in Fig. 105, and laying down the dotted circle equal in diameter to the valve travel, we lay down *a*, *b*, *a'*, *b'*, to right and left of the centre line by a distance equal to the lap—as before $1\frac{1}{8}$ inch. The full gear leads being negative, the points *c*, *d*, *c'*, *d'* will come inside of *a*, *b*, *a'*, *b'*, instead of outside, as in Fig. 105. Assuming a full forward gear negative lead of $\frac{1}{8}$ inch, *c* and *c'* are laid down with a horizontal distance of $\frac{1}{8}$ inch inside of *a* and *a'*, and similarly assuming a full backing gear negative lead of $\frac{1}{4}$ inch, *d* and *d'* are laid down with a horizontal distance of $\frac{1}{4}$ inch inside of *b* and *b'*. From these points *c*, *d*, *c' d'* the dotted link positions for the two crank centres are located, showing the midgear travel to be *e f*, which by measurement is found to be $2\frac{3}{4}$ inches. Dividing this by two, as before, and subtracting the lap *o i* or *o n*, the midgear lead *i f* or *e n* is found to equal $\frac{2\frac{3}{4}}{2} - 1\frac{1}{8} = \frac{1}{4}$ inch. Repeating the construction of Fig. 106 in 108, remembering that the full gear leads *b c* and *b' c'* are negative, gives the arc *f e g* for the line of travel for the equivalent shifting eccentric, from which the arc *e' f'* is easily located, as was done in Fig. 106. Drawing the crank line *O k* for one-third stroke as before, and the lap circle tangent to it, gives at once the lead, port opening, travel, release, and compression for one-third cut-off in the modified gear. The action in the full gear is also shown, the lead *r s* being

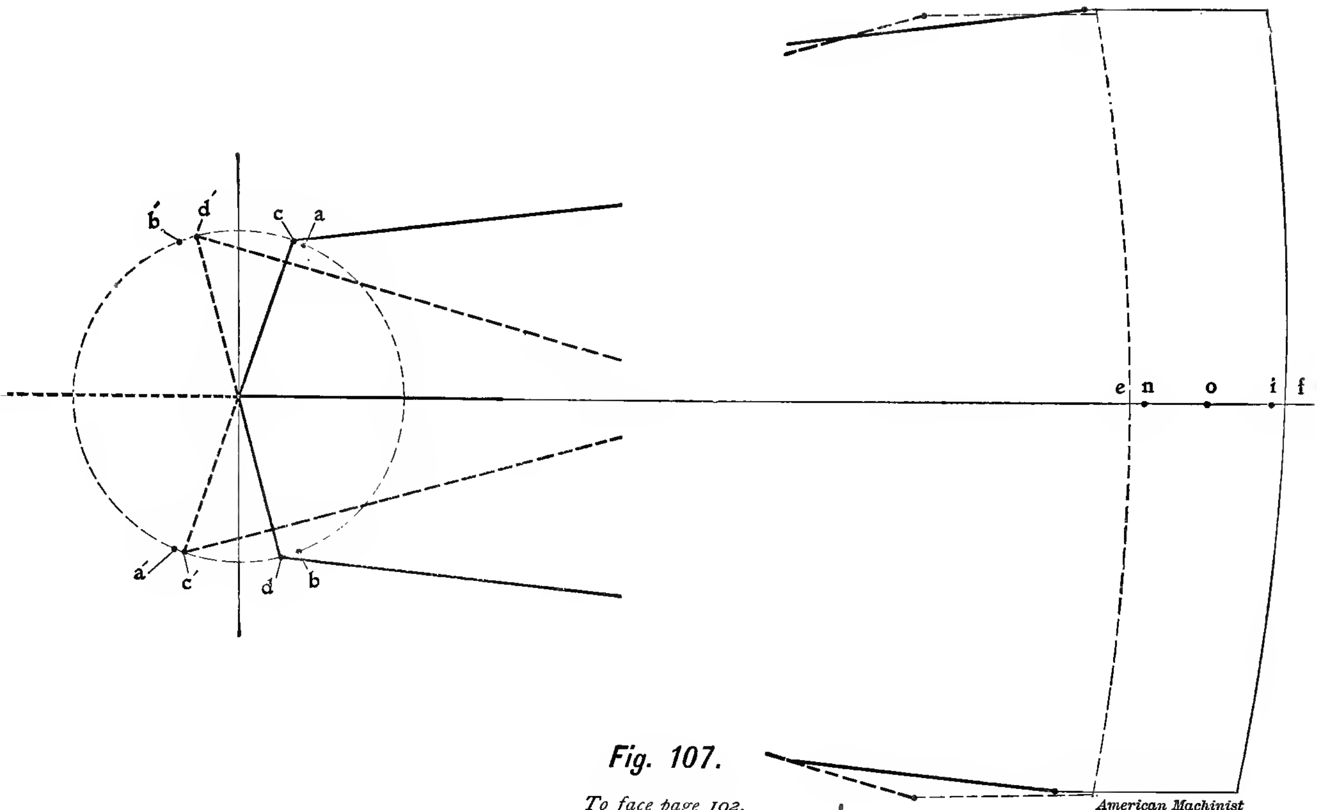

Fig. 107.

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shown to be negative by the lap circle going below the horizontal centre line. Figs. 106 and 108 are both repeated in Fig. 109, the former in full and the

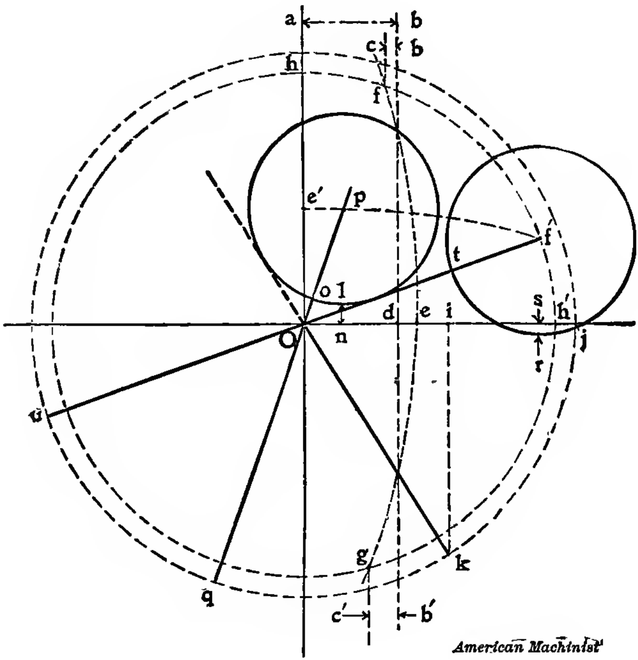

Fig. 108.

$a\,b$ = lap.
$b\,c = r\,s$ = full-gear forward lead—negative.
$b'\,c'$ = full-gear backward lead—negative.
$d\,e = l\,n$ = mid-gear lead.

latter in dotted lines, in order that the effects of the modified setting may be more easily compared. It will be seen that for the one-third cut-off, the lead and

port opening are both reduced, while the release and compression are both made later. Similar changes are also seen in the full gear and additional lap cir-

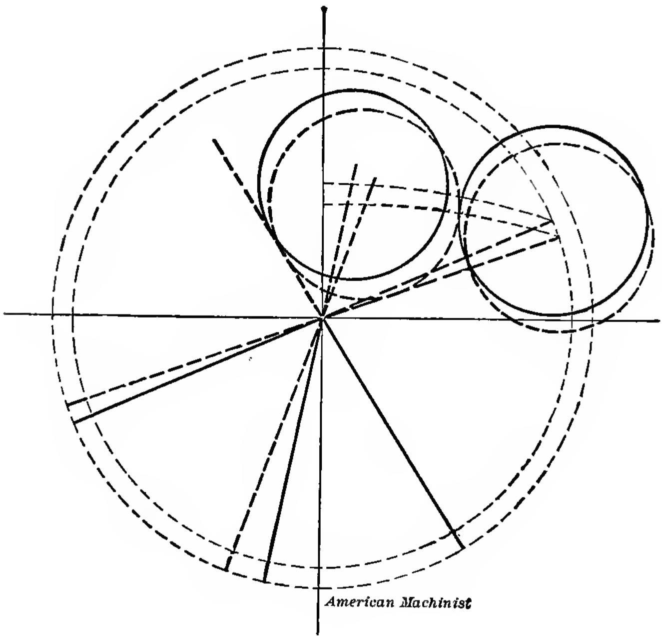

Fig. 109.

cles for other points of cut-off would show similar changes.

Link charts Nos. 3 and 4 give the records of two locomotives having the modified valve setting.

LINK CHART NO. 3.

Schenectady Locomotive Works,
———, 1897.

Built for the Northern Pacific Ry. Engine No. 4542. Size of cylinders 20″ x 26″. Size of driving wheels, 69″.

Lead.		Valve Opens.		Cut Off.		Remarks.
Forward Stroke.	Rearward Stroke.	Forward Stroke	Rearward Stroke.	Forward Stroke.	Rearward Stroke.	
$\frac{1}{8}$″ blind	$\frac{1}{8}$″ blind	$1\frac{7}{8}$″	$1\frac{7}{8}$″	$22\frac{9}{16}$″	$22\frac{5}{8}$″	Slip $1\frac{5}{8}$″.
$\frac{1}{32}$″ blind	$\frac{1}{32}$″ blind	$1\frac{7}{16}$″	$1\frac{7}{16}$″	21″	21″	
$\frac{1}{32}$″ lead	$\frac{1}{32}$″ lead	$1\frac{1}{18}$″	$1\frac{1}{16}$″	19″	19″	
$\frac{3}{32}$″	$\frac{3}{32}$″	$\frac{23}{32}$″	$\frac{23}{32}$″	16″	16″	
$\frac{1}{8}$″	$\frac{1}{8}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	13″	$13\frac{1}{8}$″	
$\frac{9}{64}$″	$\frac{9}{64}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	10″	10″	Slip $1\frac{1}{8}$″.
$\frac{5}{32}$″ S.	$\frac{5}{32}$″ S.	$\frac{5}{16}$″	$\frac{5}{16}$″	8″	8″	
$\frac{5}{32}$″	$\frac{5}{32}$″	$\frac{1}{4}$″	$\frac{1}{4}$″	6″	6″	
$\frac{5}{32}$″ F.	$\frac{5}{32}$″ F.	$\frac{7}{32}$″	$\frac{7}{32}$″	4″	$4\frac{1}{18}$″	

Passenger service, six coupled wheels. Lap, $1\frac{1}{8}$″.
Length of main rods, $126\frac{1}{2}$″. Travel, 6″.
Radius of Link, 40″. Allen valve.

LINK CHART, NO. 4.

Schenectady Locomotive Works,
Sept. 18, 1895.

Built for the Chicago and Northwestern R. R. Engine No. 4337. Size of cylinders, 19″ x 24″. Size driving wheels, 75″.

LEAD.		VALVE OPENS.		CUT OFF.		REMARKS.
Forward Stroke.	Rearward Stroke.	Forward Stroke	Rearward Stroke.	Forward Stroke.	Rearward Stroke.	Offset $\frac{23}{32}$″
$\frac{3}{16}$″ blind	$\frac{3}{16}$″ blind	$1\frac{3}{4}$″	$1\frac{3}{4}$″	$20\frac{5}{16}$″	$20\frac{1}{2}$″	Slip $\frac{3}{4}$″.
$\frac{3}{32}$″ blind	$\frac{3}{32}$″ blind	$1\frac{1}{2}$″	$1\frac{1}{2}$″	20″	$20\frac{1}{8}$″	
$\frac{1}{32}$″ blind	$\frac{1}{32}$″ blind	$1\frac{1}{4}$″	$1\frac{1}{4}$″	18″	$18\frac{1}{8}$″	
line&line	line&line	1″	1″	16″	$16\frac{1}{16}$″	
$\frac{3}{32}$″ lead	$\frac{3}{32}$″ lead	$\frac{3}{4}$″ S.	$\frac{3}{4}$″	14″	14″	
$\frac{5}{32}$″	$\frac{5}{32}$″	$\frac{3}{16}$″	$\frac{3}{16}$″	12″	12″	Slip $\frac{5}{8}$″.
$\frac{3}{16}$″	$\frac{3}{16}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	10″	10″	
$\frac{7}{32}$″	$\frac{7}{32}$″	$\frac{7}{16}$″	$\frac{7}{16}$″	8″	8″	
$\frac{1}{4}$″	$\frac{1}{4}$″	$\frac{3}{8}$″	$\frac{3}{8}$″	6″	$6\frac{1}{16}$″	

Passenger service, four coupled wheels. Lap, $1\frac{1}{4}$″.
Length of main rods, $91\frac{1}{2}$″. Travel, 6″.
Radius of Link, $59\frac{1}{2}$″. Allen valve.

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